SECOND EDITION

VOLUME 1

Surface Production Operations

Design of Oil-Handling Systems and Facilities

Ken Arnold
Maurice Stewart

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Contents

Acknowledgments

It was Maurice Stewart's idea to take the lecture notes that I had developed for my course at the University of Houston, add some material he developed for his lecture notes at Tulane University, and write this book. In addition, Maurice was responsible for first drafts of several chapters, editorial review, and comment on the finished work, and he aided greatly in developing many of the illustrations.

There are two core themes to my lecture notes and this book: (1) engineers must be aware of the first principle basis for their design procedures if they are to exercise the correct judgment in choosing between alternatives, and (2) the mystery of process vessel design can be removed and design equations unified using a drop size distribution and analysis technique that others have developed before me and I merely extended to other situations.

I am indebted to professors Robert McGuire and Richard White of Cornell University for convincing an impressionable undergraduate of the importance of the first theme. With this understanding I have spent much of my working life trying to explain observed phenomena and published answers in the field of production facility design. In this effort I have been fortunate to have worked for two of the best companies in their respective industries, Shell Oil Company and Paragon Engineering Services, Inc. Both have given me the opportunity and resources to continue to pursue this goal and apply the ideas and concepts presented in this book to real situations.

I am indebted to several colleagues within both Paragon and Shell who have aided, instructed, critiqued, and provided me with hours of argument.

For the second edition, I would like to thank the following Paragon engineers who each revised a chapter: Eric Barron, Jim Cullen, Fernando De La Fuente, Robert Ferguson, Mike Hale, Sandeep Khurana, Kevin

Mara, Matt McKinstry, Carl Sikes, Mary Thro, Kirk Trascher, and Mike Whitworth. I would also like to thank David Arnold for pulling it all together at the end.

Ken Arnold, P.E.
Houston, Texas

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Finally, I would like to express a special thanks to and dedicate this text to my son, Chad, who deserved more than a part-time father during the preparation of this text.

Maurice I. Stewart, Ph.D., P.E.
Baton Rouge, Louisiana

Preface

As teachers of production facility design courses in petroleum engineering programs, we both realized there was no single source that could be used as a text in this field. We found ourselves reproducing pages from catalogues, reports, projects we had done, etc., to provide our students with the basic information they needed to understand the lectures and carry out their assignments. Of more importance, the material that did exist usually contained nomographs, charts, and rules of thumb that had no reference to the basic theories and underlying assumptions upon which they were based or, worse, had misleading or even false statements concerning these principles.

This text, which covers about one semester's work, presents the basic concepts and techniques necessary to design, specify, and manage oil field surface production facilities. It provides a clear understanding of the equipment and processes used in common separation and oil and water treating systems, as well as the selection of piping and pumping systems. We hope this will enable you to develop a "feel" for the important parameters of designing and operating a production facility. We also wish the reader to understand the uncertainties and assumptions inherent in designing and using the equipment in these systems and the limitations, advantages, and disadvantages associated with their use.

We strongly believe that there is an engineering discipline and science to production facility design. If someone is going to be taught to apply this science intelligently, the underlying assumptions and the engineering discipline must be understood. In developing our lecture notes we structured them around derivations of design equations from first principles so that our students could see the assumptions that were made and their importance. Wherever a rule of thumb must be applied, we have attempted to explain at least qualitatively why it was reasonable and under what conditions it might be altered. Some of the material is by necessity presented as nothing more than an educated guess to be used only if no other

information in the form of laboratory studies or field experience exists. These points are clearly stated in the text. It is hoped that by publishing these thoughts we will stimulate others to publish their experiences and rules of thumb, and help stimulate some much-needed research.

Some of our students have no background in production facility design other than what they have learned in the introductory petroleum engineering courses. For this reason, we have found by trial and error that it is best to start with an overview explaining the goals of the facility with pictures of the equipment. We then discuss how the equipment is put together into a process system before explaining process calculations and equipment designing procedures. We have experimented with discussing the process system at the end of the course, but have found that the interrelationship of the various pieces of equipment confuses the student. Therefore, while there is some repetition caused by the order of the chapters, experience shows us that, in the final analysis, the present order provides the student with a clearer understanding of the concepts.

The order chosen for the book has the side benefit of allowing the instructor to assign a project at the start of the course and have the student take it another step forward as each segment is completed. An example project is included in Appendix A. As there are many correct answers in facility process and equipment design, no two projects will be identical, but the student should be able to defend his selection. One of us ends the semester by having each student defend his project in an oral presentation.

The more experienced design engineer may wish to just skim the first three chapters. We have tried to make this an easier reference book to use by visually separating the derivations from the rest of the text so that the design equations and important assumptions stand out more clearly. Where needed, we have summarized each section with a design procedure or an example calculation.

This volume focuses on areas that primarily concern oil-handling facilities. These topics are not adequately addressed in the literature, and handling oil and water is much more of an art and less of a science than handling gas. The book does not cover gas dehydration and treating, gas compression, electric generators and distribution, material selection and corrosion control, chemical treatment, welding, water purification and steam generation equipment, instrument specification and system design,

foundations, platforms, buildings, and many other topics that fall under the responsibility of a surface facility or construction engineer.

The final chapter on project management may appear a little out of place. However, it is our experience that a large part of the production facility engineer's job is project management. It would be a shame if the students taking our course eventually became managers of production facility engineers and were never exposed to the concepts contained in this chapter. One of the most common comments we hear from engineers in this field is that the petroleum engineers who supervise them have no concept of the problems associated with managing one of these projects.

Throughout the book we have attempted to concentrate on what we perceive to be modern and common practices. Although we have either personally been involved in the design of facilities all over the world or have people in our organizations who have done so, undoubtedly we are influenced by our own experience and prejudices. We apologize if we have left something out or have expressed opinions on equipment types that differ from your experiences. We have learned much from our students' comments on such matters and would appreciate receiving yours for future revisions/editions.

Ken Arnold, P.E.
Houston, Texas

Maurice I. Stewart, Ph.D., P.E.
Baton Rouge, Louisiana

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CHAPTER

1

The Production Facility

INTRODUCTION

The job of a production facility is to separate the well stream into three components, typically called "phases" (oil, gas, and water), and process these phases into some marketable product(s) or dispose of them in an environmentally acceptable manner. In mechanical devices called "separators" gas is flashed from the liquids and "free water" is separated from the oil. These steps remove enough light hydrocarbons to produce a stable crude oil with the volatility (vapor pressure) to meet sales criteria. Figures 1-1 and 1-2 show typical separators used to separate gas from liquid or water from oil. Separators can be either horizontal or vertical in configuration.

The gas that is separated must be compressed and treated for sales. Compression is typically done by engine-driven reciprocating compressors, Figure 1-3. In large facilities or in booster service, turbine-driven centrifugal compressors, such as that shown in Figure 1-4, are used. Large integral reciprocating compressors are also used, Figure 1-5.

Usually, the separated gas is saturated with water vapor and must be dehydrated to an acceptable level (normally less than 7 lb/MMscf). Usually this is done in a glycol dehydrator, such as that shown in Figure 1-6.

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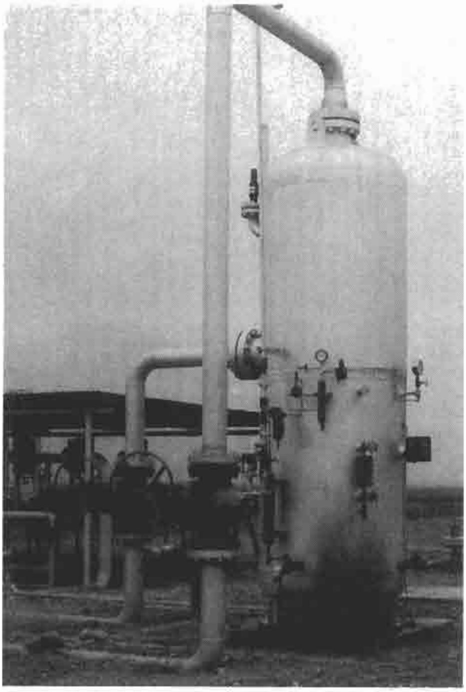

Figure 1-1. A typical vertical two-phase separator at a land location. The inlet comes in the left side, gas comes off the top, and liquid leaves the bottom right side of the separator.

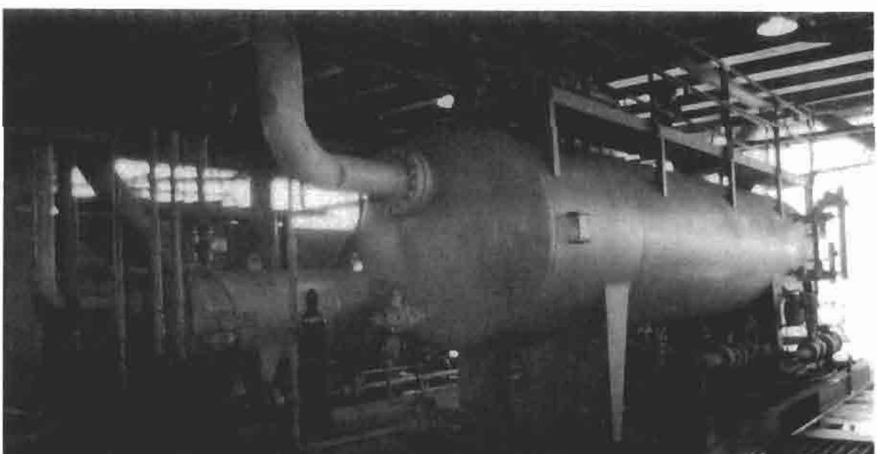

Figure 1-2. A typical horizontal separator on an offshore platform showing the inlet side. Note the drain valves at various points along the bottom and the access platform along the top.

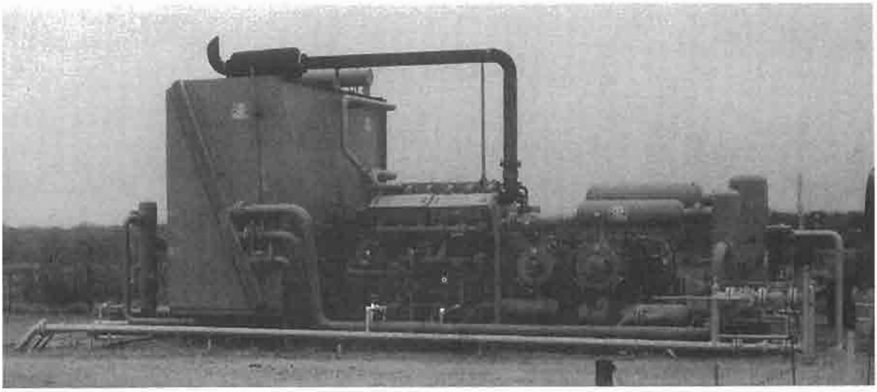

Figure 1-3. An engine-driven compressor package. The inlet and interstage scrubbers (separators) are at the right. The gas is routed through pulsation bottles to gas cylinders and then to the cooler on the left end of the package. The engine that drives the compressor cylinders is located to the right of the box-like cooler.

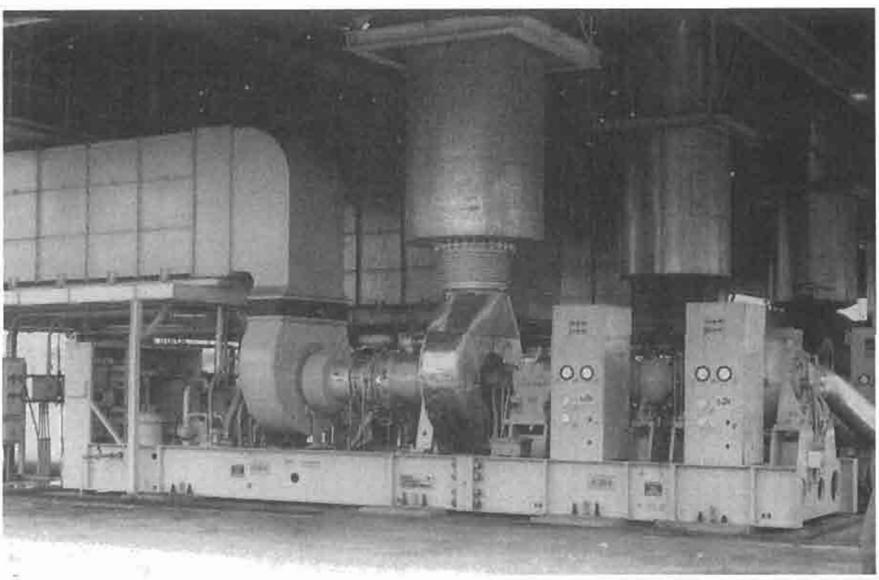

Figure 1-4. A turbine-driven centrifugal compressor. The turbine draws air in from the large duct on the left. This is mixed with fuel and ignited. The jet of gas thus created causes the turbine blades to turn at high speed before being exhausted vertically upward through the large cylindrical duct. The turbine shaft drives the two centrifugal compressors, which are located behind the control cabinets on the right end of the skid.

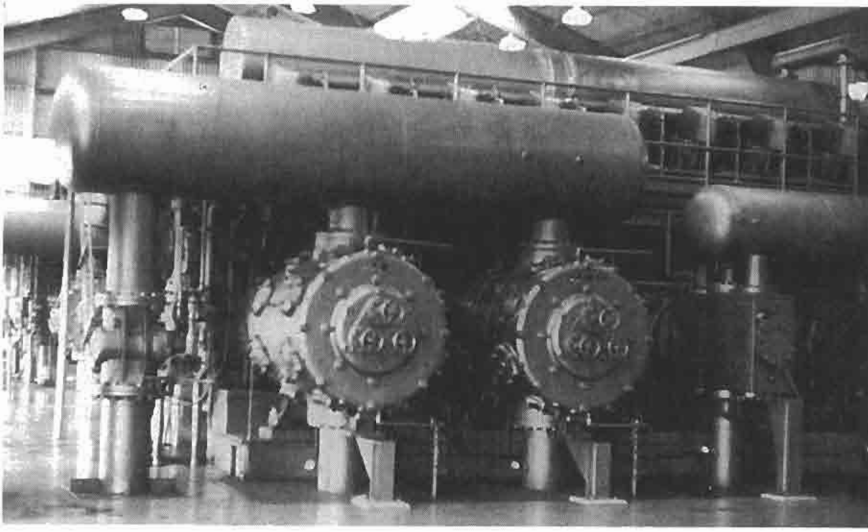

Figure 1-5. A 5500-hp integral reciprocating compressor. The sixteen power cylinders located at the top of the unit (eight on each side) drive a crankshaft that is directly coupled to the horizontal compressor cylinders facing the camera. Large cylindrical "bottles" mounted above and below the compressor cylinders filter out acoustical pulsations in the gas being compressed.

(text continued from page 1)

Dry glycol is pumped to the large vertical contact tower where it strips the gas of its water vapor. The wet glycol then flows through a separator to the large horizontal reboiler where it is heated and the water boiled off as steam.

In some locations it may be necessary to remove the heavier hydrocarbons to lower the hydrocarbon dew point. Contaminants such as H_2S and CO_2 may be present at levels higher than those acceptable to the gas purchaser. If this is the case, then additional equipment will be necessary to "sweeten" the gas.

The oil and emulsion from the separators must be treated to remove water. Most oil contracts specify a maximum percent of basic sediment and water (BS and W) that can be in the crude. This will typically vary from 0.5% to 3% depending on location. Some refineries have a limit on salt content in the crude, which may require several stages of dilution with fresh water and subsequent treating to remove the water. Typical salt limits are 10 to 25 pounds of salt per thousand barrels.

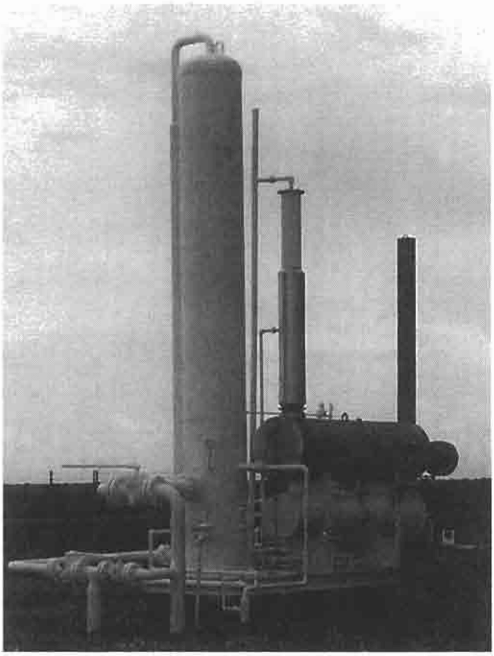

Figure 1-6. A small glycol gas dehydration system. The large vertical vessel on the left is the contact tower where "dry" glycol contacts the gas and absorbs water vapor. The upper horizontal vessel is the "reboiler" or "reconcentrator" where the wet glycol is heated, boiling off the water that exits the vertical pipe coming off the top just behind the contact tower. The lower horizontal vessel serves as a surge tank.

Figures 1-7 and 1-8 are typical direct-fired heater-treaters that are used for removing water from the oil and emulsion being treated. These can be either horizontal or vertical in configuration and are distinguished by the fire tube, air intakes, and exhausts that are clearly visible. Treaters can be built without fire tubes, which makes them look very much like separators. Oil treating can also be done by settling or in gunbarrel tanks, which have either external or internal gas boots. A gunbarrel tank with an internal gas boot is shown in Figure 1-9.

Production facilities must also accommodate accurate measuring and sampling of the crude oil. This can be done automatically with a Lease Automatic Custody Transfer (LACT) unit or by gauging in a calibrated tank. Figure 1-10 shows a typical LACT unit.

The water that is produced with crude oil can be disposed of overboard in most offshore areas, or evaporated from pits in some locations onshore. Usually, it is injected into disposal wells or used for waterflooding. In any case, water from the separators must be treated to remove small quantities of produced oil. If the water is to be injected into a disposal well, facilities may be required to filter solid particles from it.

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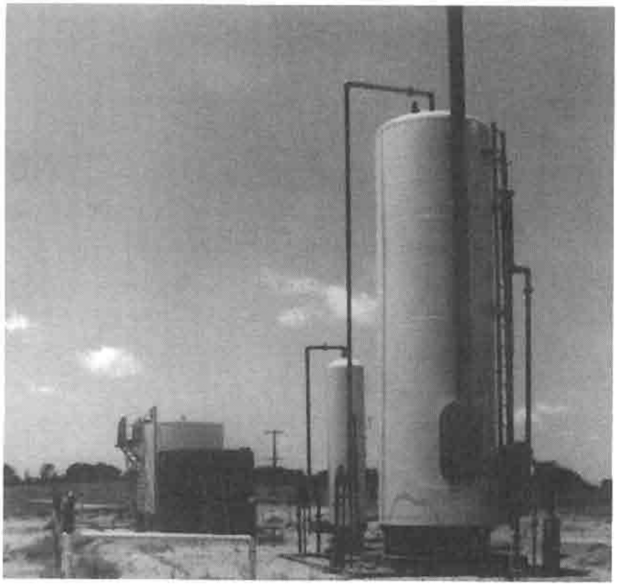

Figure 1-7. A vertical heater treater. The emulsion to be treated enters on the far side. The fire tubes (facing the camera) heat the emulsion, and oil exits near the top. Water exits the bottom through the external water leg on the right, which maintains the proper height of the interface between oil and water in the vessel. Gas exits the top. Some of the gas goes to the small "pot" at the lower right where it is scrubbed prior to being used for fuel for the burners.

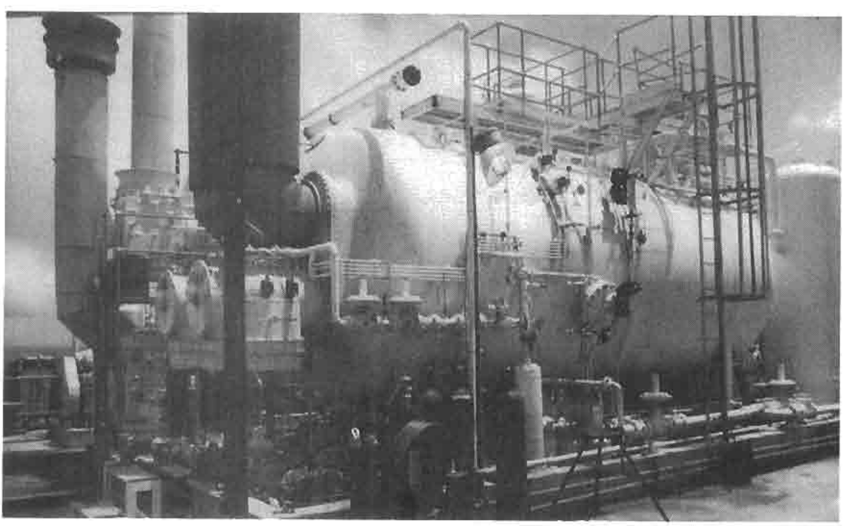

Figure 1-8. A horizontal heater treater with two burners.

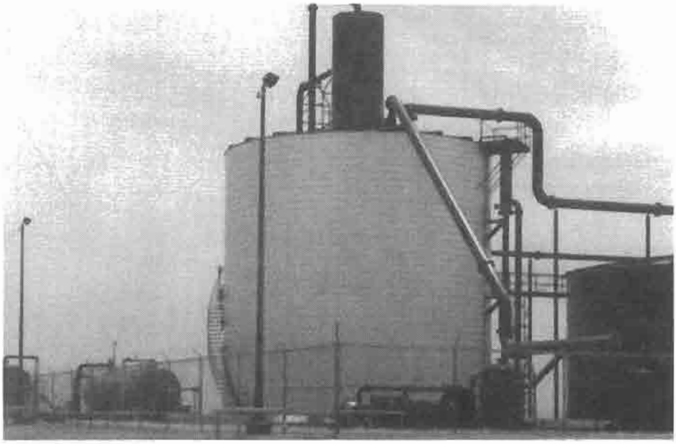

Figure 1-9. A gunbarrel tank for treating oil. The emulsion enters the "gas boot" on top where gas is liberated and then drops into the tank through a specially designed "downcomer" and spreader system. The interface between oil and water is maintained by the external water leg attached to the right side of the tank. Gas from the tank goes through the inclined pipe to a vapor recovery compressor to be salvaged for fuel use.

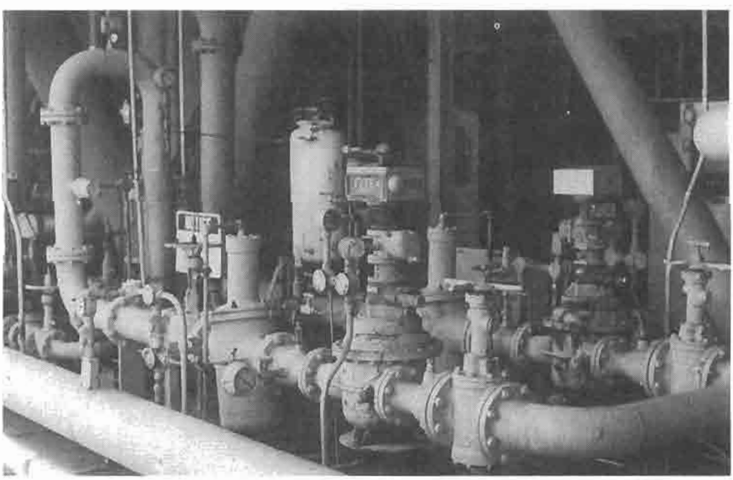

Figure 1-10. A LACT unit for custody transfer of oil. In the vertical loop on the left are BS&W probe and a sampler unit. The flow comes through a strainer with a gas eliminator on top before passing through the meter. The meter contains devices for making temperature and gravity corrections, for driving the sampler, and for integrating the meter output with that of a meter prover (not shown).

(text continued from page 5)

Water treating can be done in horizontal or vertical skimmer vessels, which look very much like separators. Water treating can also be done in one of the many proprietary designs discussed in this text such as upflow or downflow CPIs (Figure 1-11), flotation units (Figure 1-12), crossflow coalescers/separators, and skim piles. Skim tanks with and without free-flow turbulent coalescers (SP Packs) can also be used.

Any solids produced with the well stream must also be separated, cleaned, and disposed of in a manner that does not violate environmental criteria. Facilities may include sedimentation basins or tanks, hydrocyclones, filters, etc. Figure 1-13 is a typical hydrocyclone or "desander" installation.

The facility must provide for well testing and measurement so that gas, oil, and water production can be properly allocated to each well. This is necessary not only for accounting purposes but also to perform reservoir studies as the field is depleted.

The preceding paragraphs summarize the main functions of a production facility, but it is important to note that the auxiliary systems support-

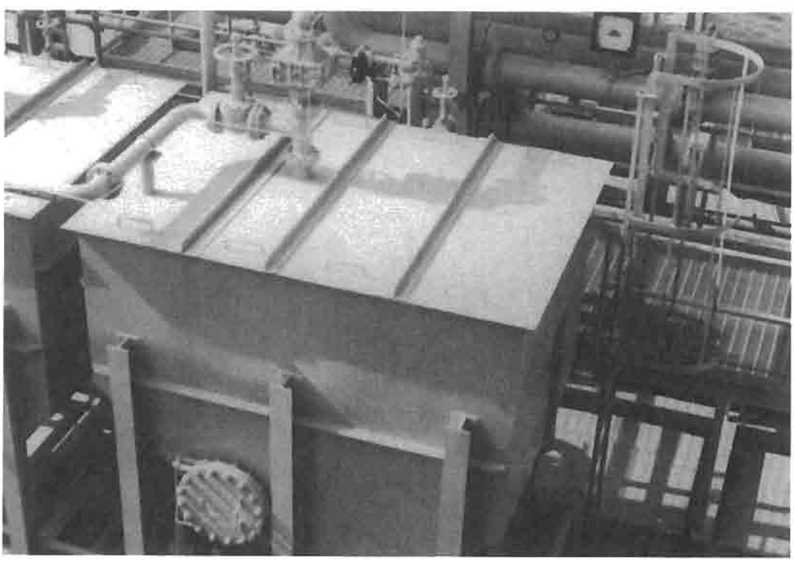

Figure 1-11. A corrugated plate interceptor (CPI) used for treating water. Note that the top plates are removable so that maintenance can be performed on the plates located internally to the unit.

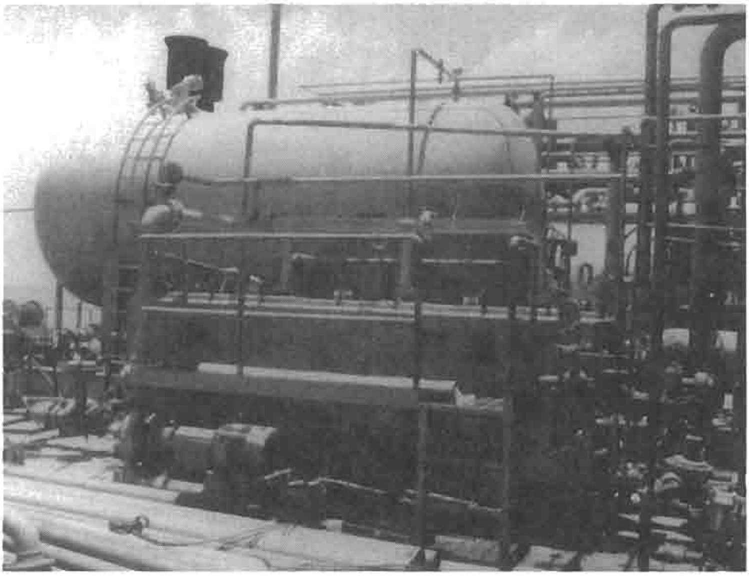

Figure 1-12. A horizontal skimmer vessel for primary separation of oil from water with a gas flotation unit for secondary treatment located in the foreground. Treated water from the flotation effluent is recycled by the pump to each of the three cells. Gas is sucked into the stream from the gas space on top of the water by a venturi and dispersed in the water by a nozzle.

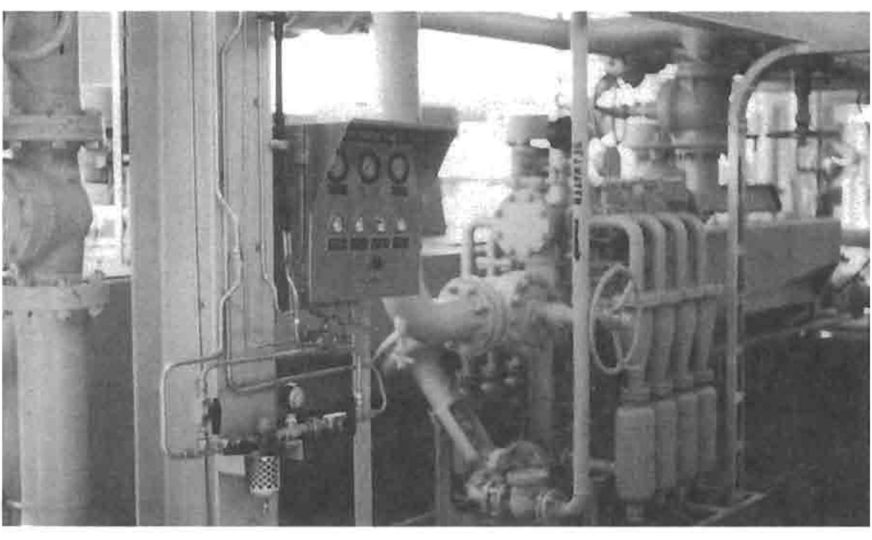

Figure 1-13. Hydrocyclone desanders used to separate sand from produced water prior to treating the water.

ing these functions often require more time and engineering effort than the production itself. These support efforts include:

1. Developing a site with roads and foundations if production is onshore, or with a platform, tanker, or some more exotic structure if production is offshore.
2. Providing utilities to enable the process to work: generating and distributing electricity; providing and treating fuel gas or diesel; providing instrument and power air; treating water for desalting or boiler feed, etc. Figure 1-14 shows a typical generator installation and Figure 1-15 shows an instrument air compressor.
3. Providing facilities for personnel, including quarters (Figure 1-16), switchgear and control rooms (Figure 1-17), workshops, cranes, sewage treatment units (Figure 1-18), drinking water (Figure 1-19), etc.
4. Providing safety systems for detecting potential hazards (Figures 1-20 and 1-21), fighting hazardous situations when they occur (Figures 1-22 and 1-23) and for personnel protection and escape (Figure 1-24).

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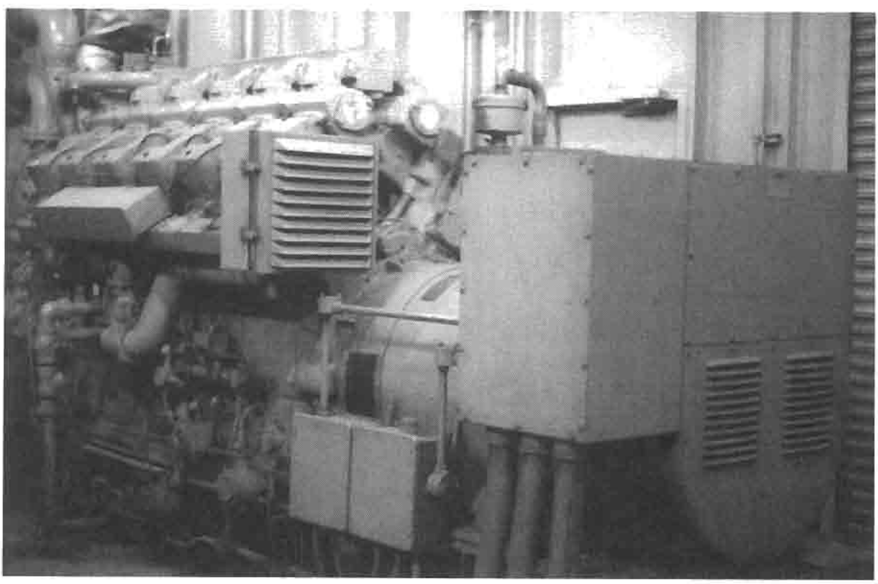


Figure 1-14. A gas-engine-driven generator located in a building on an offshore platform.

Figure 1-15. A series of three electric-motor-driven instrument air compressors. Note each one has its own cooler. A large air receiver is included to minimize the starting and stopping of the compressors and to assure an adequate supply for surges.

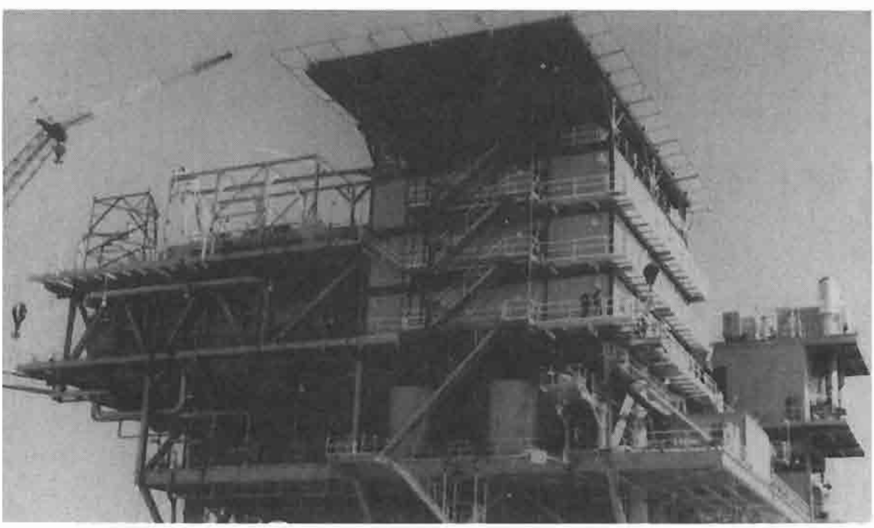


Figure 1-16. A three-story quarters building on a deck just prior to loadout for cross-ocean travel. A helideck is located on top of the quarters.

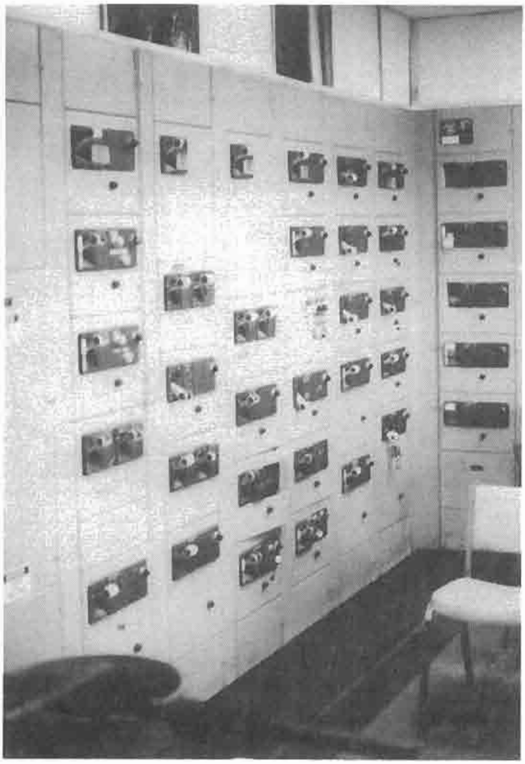


Figure 1-17. A portion of the motor control center for an offshore platform.

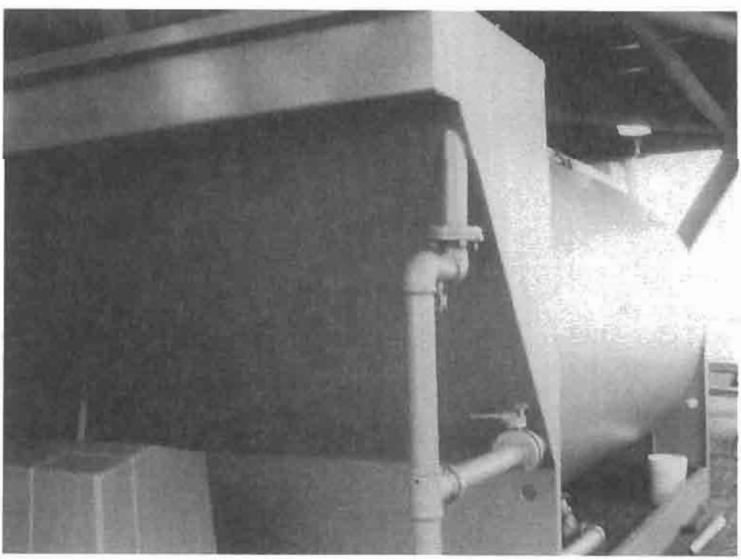


Figure 1-18. An activated sludge sewage treatment unit for an offshore platform.

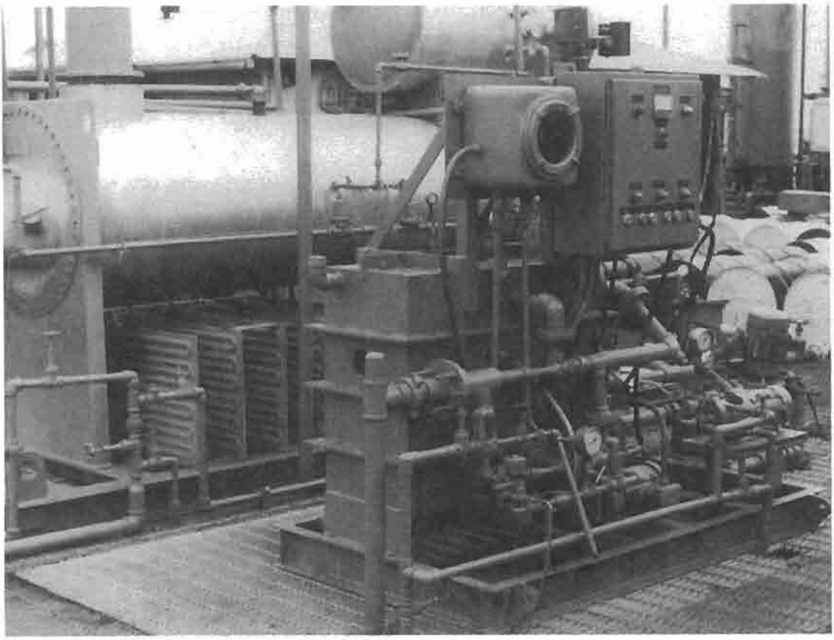


Figure 1-19. A vacuum distillation water-maker system.

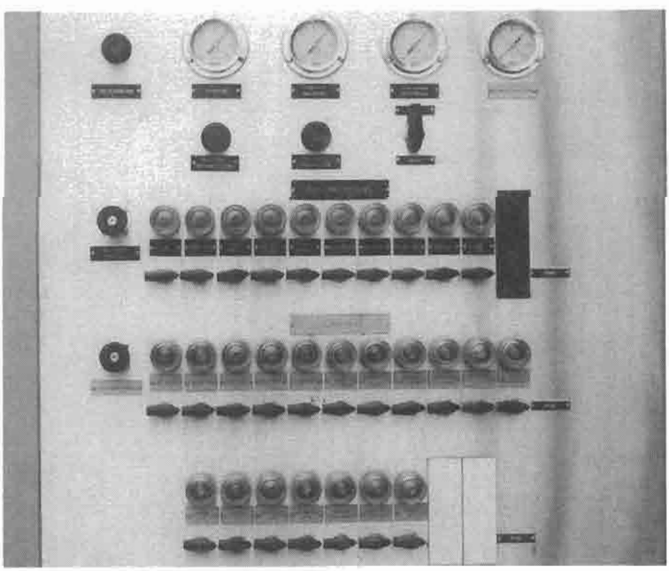


Figure 1-20. A pneumatic shut-in panel with "first-out" indication to inform the operator of which end element caused the shutdown.

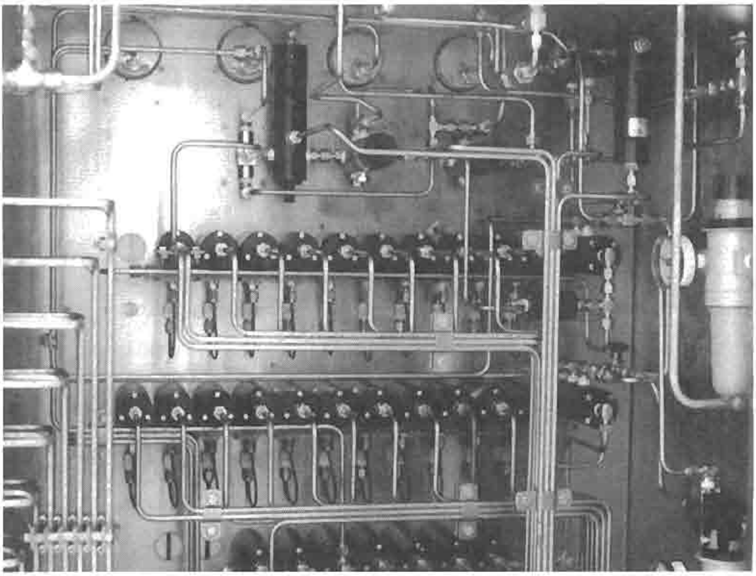


Figure 1-21. The pneumatic logic within the panel shown in Figure 1-20.

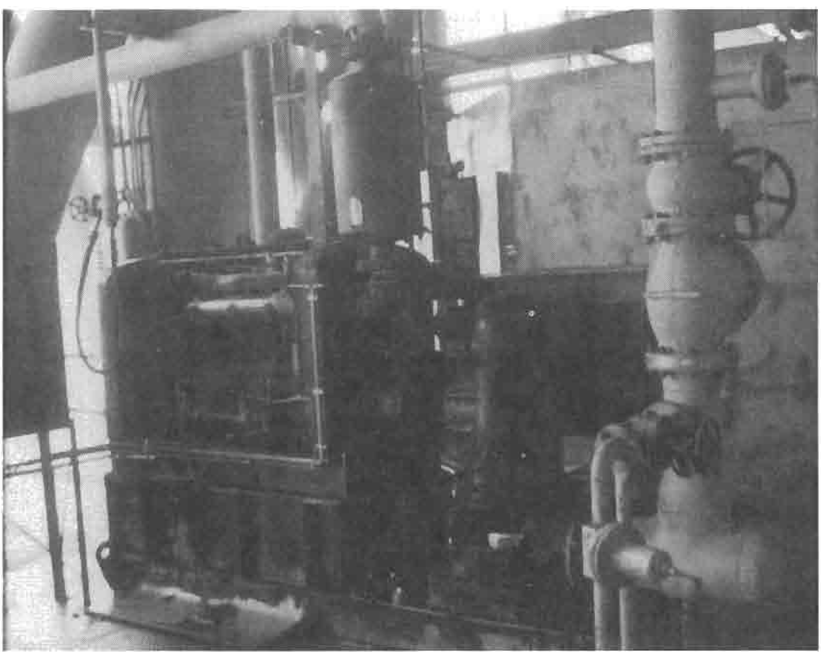


Figure 1-22. Diesel engine driven fire-fighting pump driving a vertical turbine pump through a right angle gear.

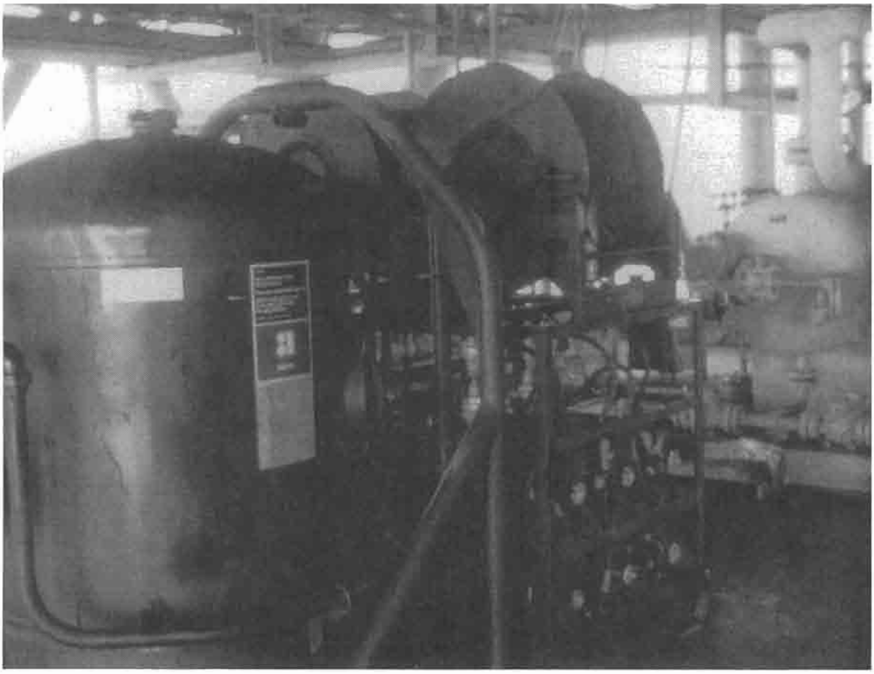


Figure 1-23. A foam fire-fighting station.

Figure 1-24. An escape capsule mounted on the lower deck of a platform. The unit contains an automatic lowering device and motor for leaving the vicinity of the platform.

(text continued from page 10)

MAKING THE EQUIPMENT WORK

The main items of process equipment have automatic instrumentation that controls the pressure and/or liquid level and sometimes temperature within the equipment. Figure 1-25 shows a typical pressure controller and control valve. In the black box (the controller) is a device that sends a signal to the actuator, which opens/closes the control valve to control pressure. Figure 1-26 shows a self-contained pressure controller, which has an internal mechanism that senses the pressure and opens/closes the valve as required.

Figure 1-27 shows two types of level controllers that use floats to monitor the level. The one on the left is an on/off switch, and the two on the right send an ever-increasing or decreasing signal as the level changes. These floats are mounted in the chambers outside the vessel. It is also possible to mount the float inside. Capacitance and inductance

Figure 1-25. A pressure control valve with pneumatic actuator and pressure controller mounted on the actuator. The control mechanism in the box senses pressure and adjusts the supply pressure to the actuator diaphragm causing the valve stem to move up and down as required.

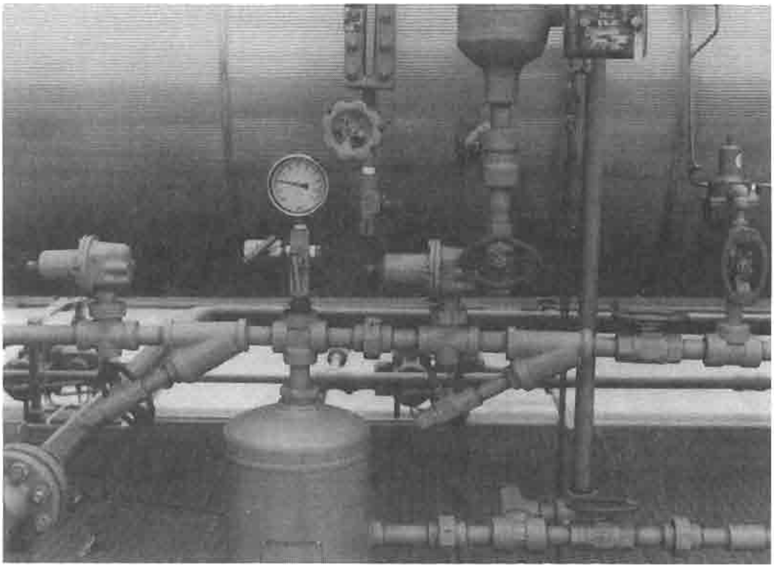


Figure 1-26. Two self-contained pressure regulators in a fuel gas piping system. An internal diaphragm and spring automatically adjust the opening in the valve to maintain pressure.

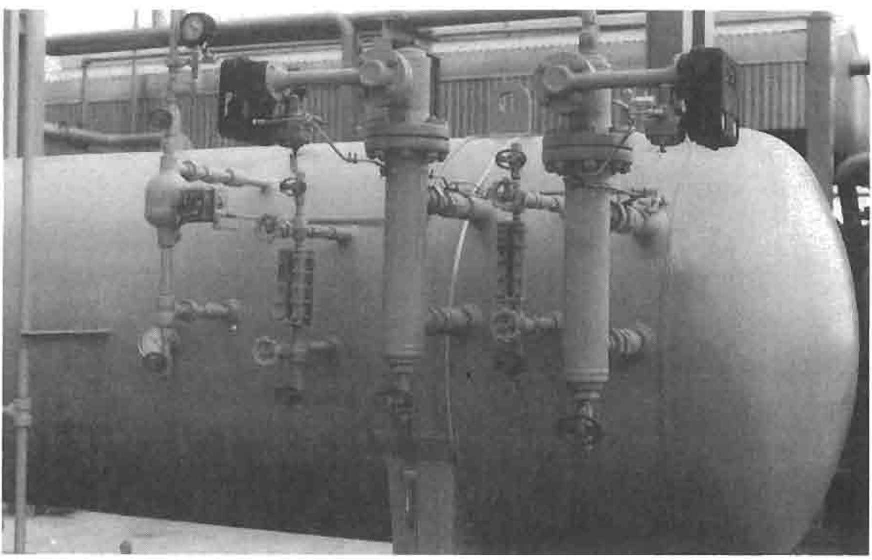


Figure 1-27. Two external level float controllers and an external float switch. The controllers on the right sense the level of fluids in the vessel. The switch on the left provides a high level alarm.

probes and pressure differential measuring devices are also commonly used to measure level.

Figure 1-28 shows a pneumatic level control valve that accepts the signal from the level controller and opens/closes to allow liquid into or out of the vessel. In older leases it is common to attach the valve to a controller float directly through a mechanical linkage. Some low-pressure installations use a lever-balanced valve such as shown in Figure 1-29. The weight on the lever is adjusted until the force it exerts to keep the valve closed is balanced by the opening force caused by the head of liquid in the vessel.

Temperature controllers send signals to control valves in the same manner as pressure and level controllers.

FACILITY TYPES

It is very difficult to classify production facilities by type, because they differ due to production rates, fluid properties, sale and disposal requirements, location, and operator preference. Some more or less typical

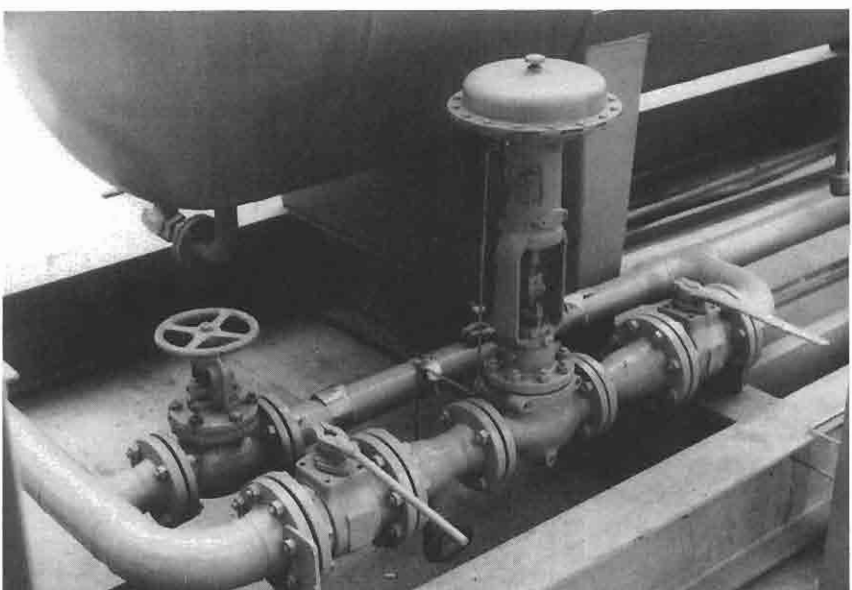


Figure 1-28. A level control valve with bypass. The signal from the controller causes the diaphragm of the actuator and thus the valve stem to move.

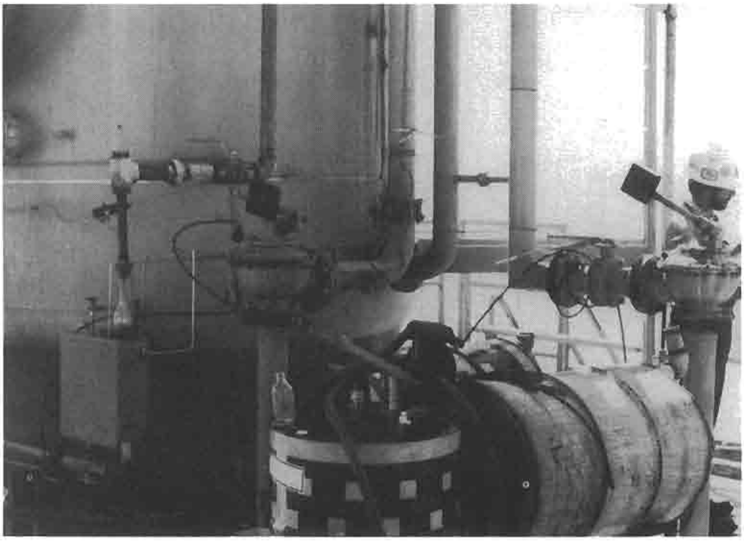


Figure 1-29. Two level-balanced liquid control valves. The position of the weight on the valve lever determines the amount of fluid column upstream of the valve necessary to force the valve to open.

cal onshore facilities are shown in Figures 1-30, 1-31, and 1-32. In cold weather areas, individual pieces of equipment could be protected as shown in Figure 1-33, or the equipment could be completely enclosed in a building such as shown in Figure 1-34.

In marsh areas the facilities can be installed on wood, concrete, or steel platforms or on steel or concrete barges, as shown in Figure 1-35. In shallow water, facilities can be installed on several different platforms connected by bridges (Figure 1-36). In deeper water it may be necessary to install all the facilities and the wells on the same platform as in Figure 1-37. Sometimes, in cold weather areas, the facilities must be enclosed as shown in Figure 1-38.

Facilities have been installed on semi-submersible floating structures, tension leg platforms, tankers (Figure 1-39) and converted jack-up drilling rigs (Figure 1-40). Figure 1-41 shows a facility installed on a man-made island.

Figure 1-30. An onshore lease facility showing vertical three-phase separator, a horizontal two-phase separator, a vertical heater treater, and two storage tanks.

Figure 1-31. An onshore central facility with a large horizontal free water knockout, and a horizontal heater treater.

Figure 1-32. A marsh facility where the equipment is elevated on concrete platforms. Note the two large vertical separators in the distance, the row of nine vertical heater treaters, and the elevated quarters building.

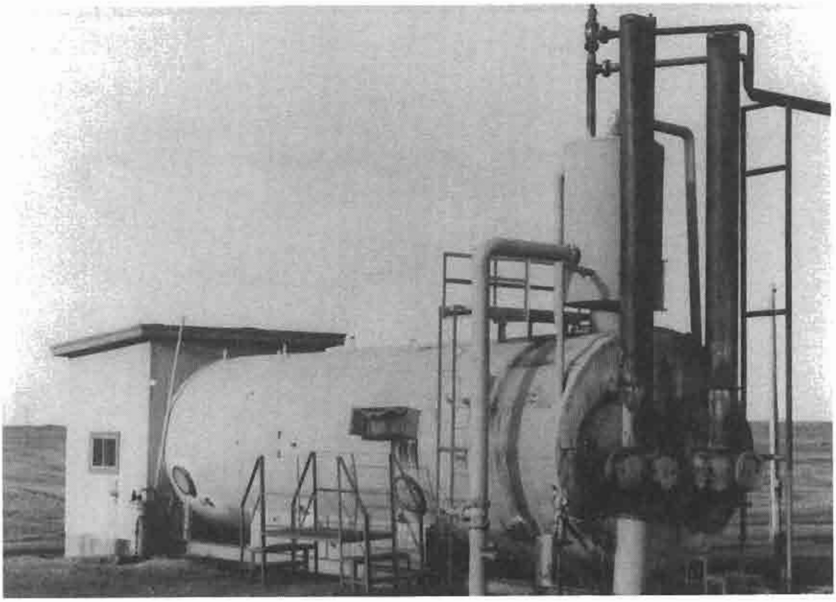


Figure 1-33. In cold weather areas it is sometimes necessary to insulate the vessels and pipe and house all controls in a building attached to the vessel.

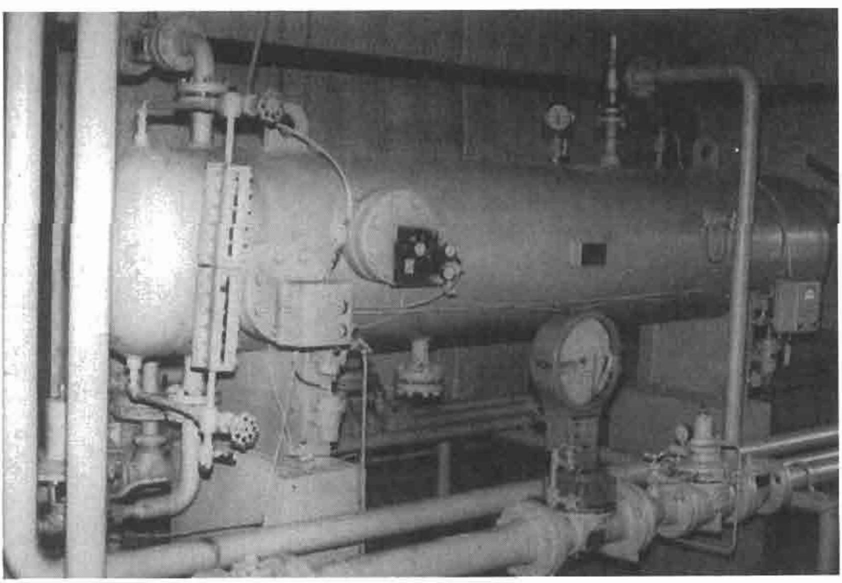


Figure 1-34. An onshore facility in Michigan where the process vessels are enclosed inside of an insulated building.

Figure 1-35. In marsh and shallow water areas it is sometimes beneficial to build the facilities on a concrete barge onshore and then sink the barge on location.

Figure 1-36. In moderate water depths it is possible to separate the quarters (on the left) and oil storage (on the right) from the rest of the equipment for safety reasons.

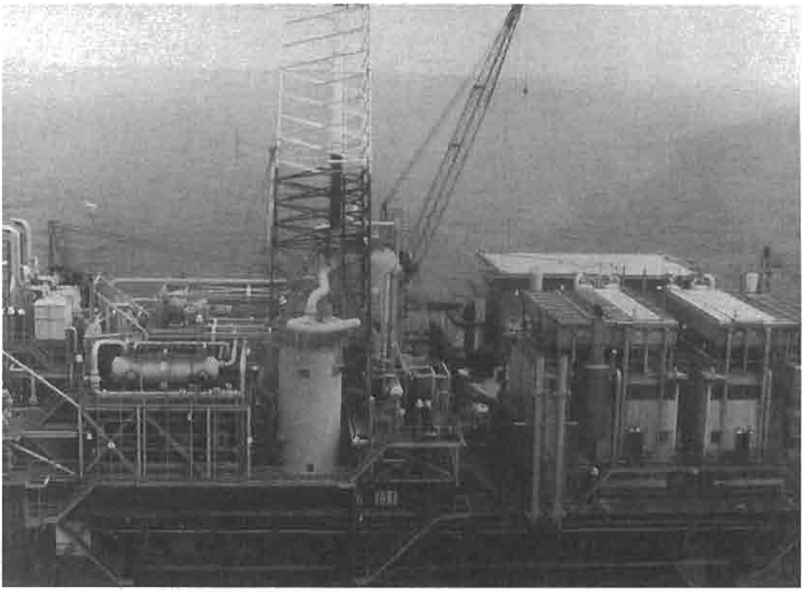


Figure 1-37. In deep waters this is not possible and the facilities can get somewhat crowded.

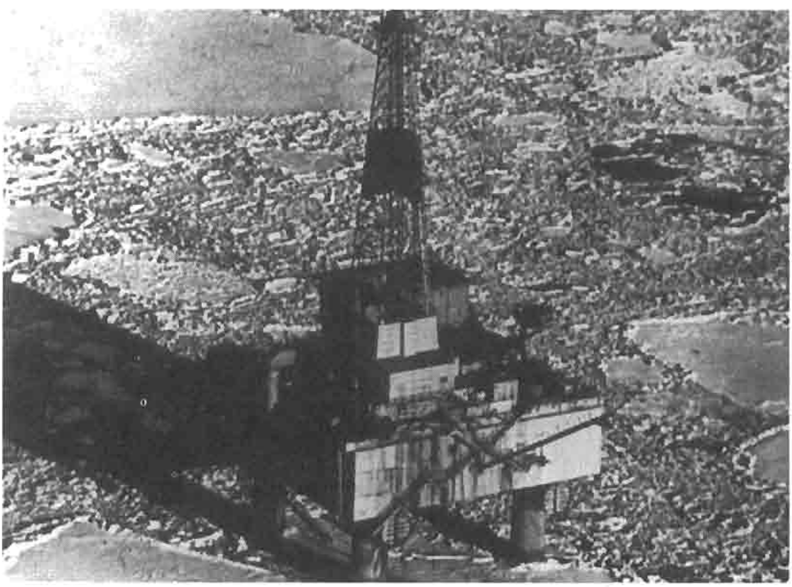


Figure 1-38. In cold weather areas such as this platform in Cook Inlet, Alaska, the facilities may be totally enclosed.

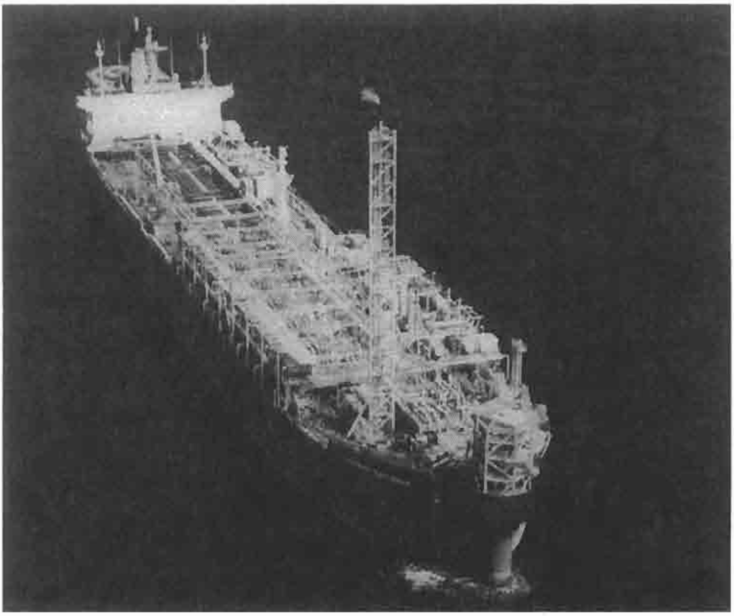


Figure 1-39. A tanker with facilities installed for a location near Thailand.

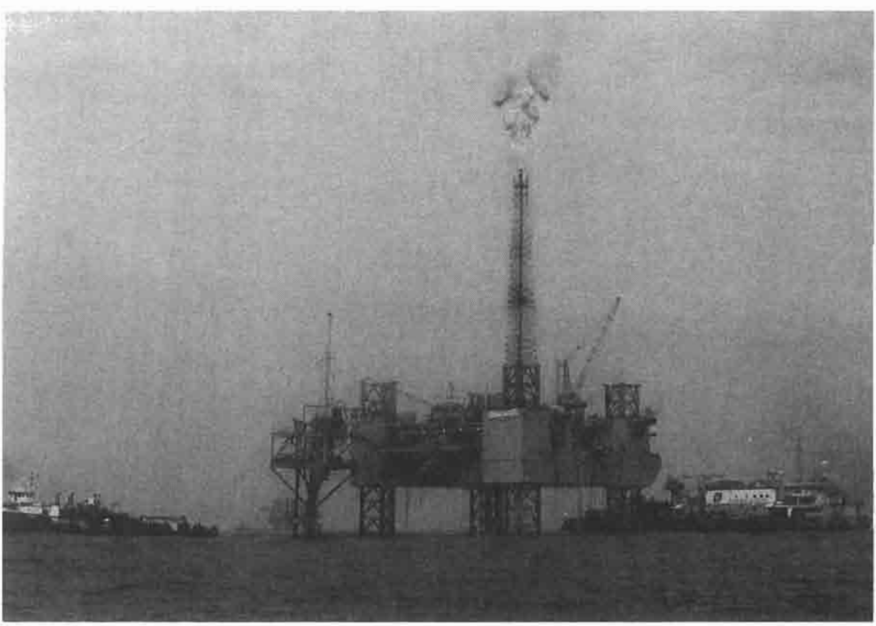


Figure 1-40. This converted jack-up rig was installed off the African coast.

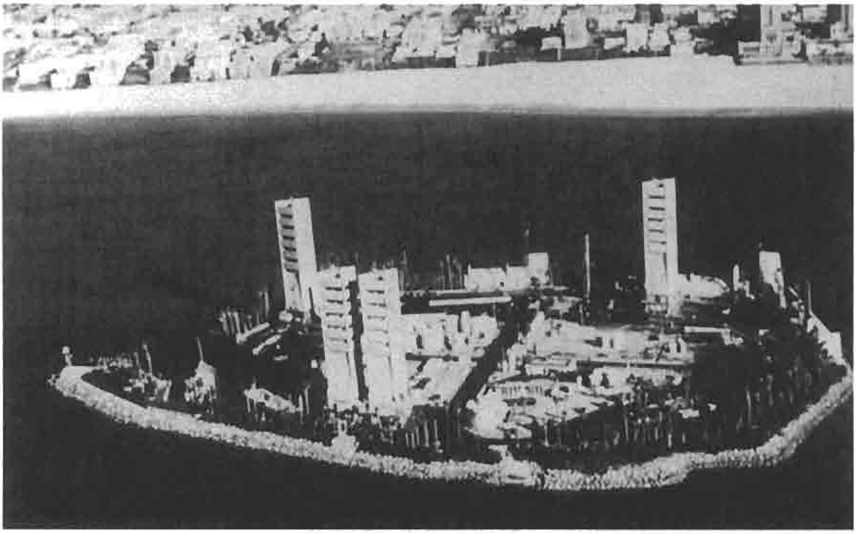


Figure 1-41. Sometimes the facilities must be decorated to meet some group's idea of what is aesthetically pleasing. This facility off California has palm trees, fake waterfalls and drilling derricks disguised as condominiums.

CHAPTER

2

Choosing a Process*

INTRODUCTION

This chapter explains how the various components are combined into a production system. The material is in no way meant to be all-inclusive. Many things must be considered in selecting components for a system and there is no substitute for experience and good engineering judgment.

A process flowsheet is used to describe the system. (Process flowsheets and their uses are described in Chapter 13.) Figure 2-1 is a typical flowsheet that will be used as an example for discussion purposes. Figure 2-2 defines many of the commonly used symbols in process flowsheets.

CONTROLLING THE PROCESS

Before discussing the process itself, it is necessary to understand how the process is controlled.

Operation of a Control Valve

Control valves are used throughout the process to control pressure, level, temperature, or flow. It is beyond the scope of this text to discuss

*Reviewed for the 1998 edition by Kevin R. Mara of Paragon Engineering Services, Inc.

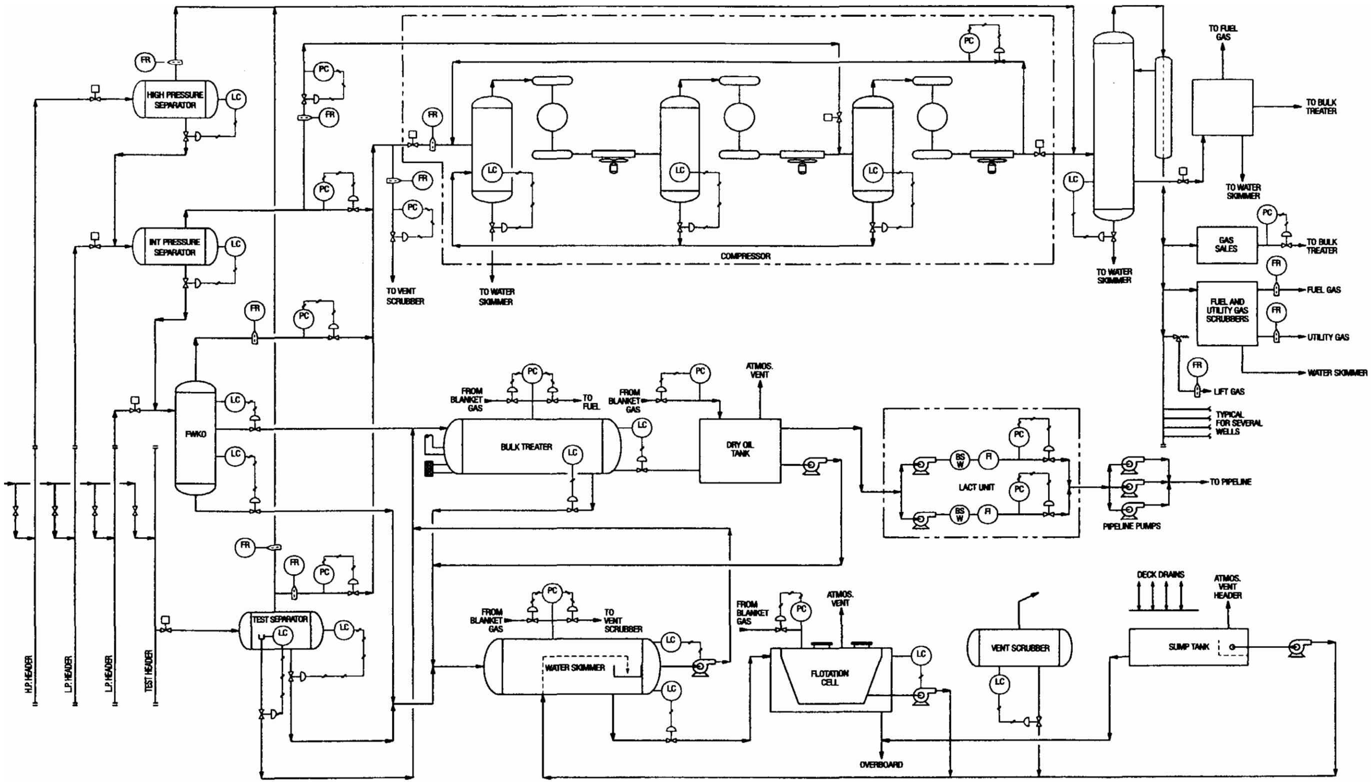


Figure 2-1. Typical flowsheet.

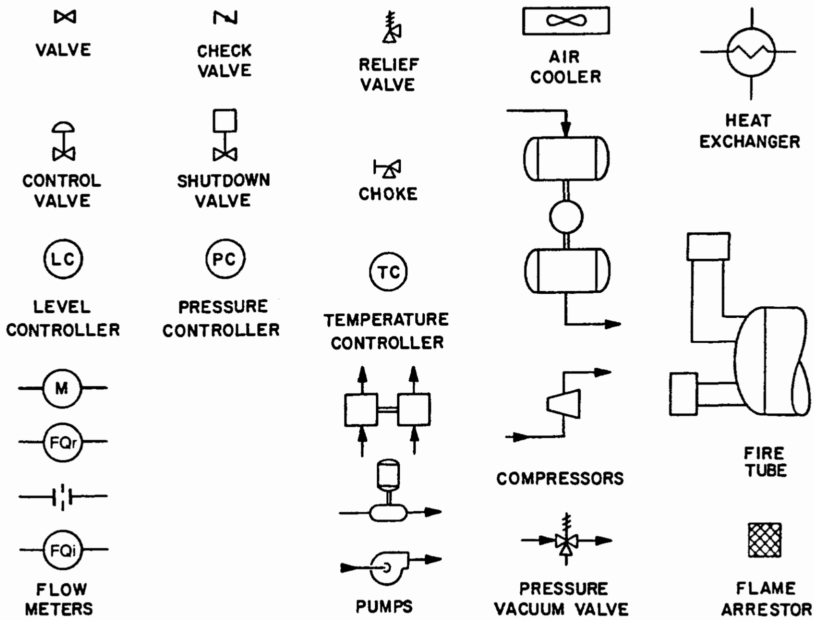


Figure 2-2. Common flowsheet symbols.

the differences between the various types of control valves and the procedures for their sizing. This section focuses primarily on the functions of this equipment. Figure 2-3 shows a very common single-port globe body control valve. All control valves have a variable opening or orifice. For a given pressure drop across the valve, the larger the orifice the greater the flow through the valve.

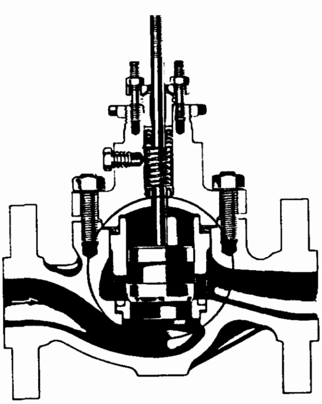


Figure 2-3. Typical single-port body control valve (*courtesy of Fisher Controls International, Inc.*).

Chokes and other flow control devices have either a fixed or variable orifice. With a fixed pressure drop across the device (i.e., with both the upstream and downstream pressure fixed by the process system) the larger the orifice the greater the flow.

In Figure 2-3 the orifice is made larger by moving the valve stem upward. This moves the plug off the seat, creating a larger annulus for flow between the seat and the plug. Similarly, the orifice is made smaller by moving the valve stem downward. The most common way to affect this motion is with a pneumatic actuator, such as that shown in Figure 2-4. Instrument air or gas applied to the actuator diaphragm overcomes a spring resistance and either moves the stem upward or downward.

The action of the actuator must be matched with the construction of the valve body to assure that the required failure mode is met. That is, if it is desirable for the valve to fail closed, then the actuator and body must be matched so that on failure of the instrument air or gas, the spring causes the stem to move in the direction that blocks flow (i.e., fully shut). This would normally be the case for most liquid control valves. If it is desirable for the valve to fail to open, as in many pressure control situations, then the spring must cause the stem to move in the fully open direction.

Pressure Control

The hydrocarbon fluid produced from a well is made up of many components ranging from methane, the lightest and most gaseous hydrocarbon, to some very heavy and complex hydrocarbon compounds. Because of this, whenever there is a drop in fluid pressure, gas is liberated. Therefore, pressure control is important.

Figure 2-4. Typical pneumatic actuator (*courtesy of Fisher Controls International, Inc.*).

The most common method of controlling pressure is with a pressure controller and a backpressure control valve. The pressure controller senses the pressure in the vapor space of the pressure vessel or tank. By regulating the amount of gas leaving the vapor space, the backpressure control valve maintains the desired pressure in the vessel. If too much gas is released, the number of molecules of gas in the vapor space decreases, and thus the pressure in the vessel decreases. If insufficient gas is released, the number of molecules of gas in the vapor space increases, and thus the pressure in the vessel increases.

In most instances, there will be enough gas separated or "flashed" from the liquid to allow the pressure controller to compensate for changes in liquid level, temperature, etc., which would cause a change in the number of molecules of gas required to fill the vapor space at a given pressure. However, under some conditions where there has been only a small pressure drop from the upstream vessel, or where the crude GOR (gas/oil ratio) is low, it may be necessary to add gas to the vessel to maintain pressure control at all times. This is called "make-up" or "blanket" gas. Gas from a pressure source higher than the desired control pressure is routed to the vessel by a pressure controller that senses the vessel pressure automatically, allowing either more or less gas to enter the vessel as required.

Level Control

It is also necessary to control the gas/liquid interface or the oil/water interface in process equipment. This is done with a level controller and liquid dump valve. The most common form of level controller is a float, although electronic sensing devices can also be used. If the level begins to rise, the controller signals the liquid dump valve to open and allow liquid to leave the vessel. If the level in the vessel begins to fall, the controller signals the liquid dump valve to close and decrease the flow of liquid from the vessel. In this manner the liquid dump valve is constantly adjusting its opening to assure that the rate of liquid flowing into the vessel is matched by the rate out of the vessel.

Temperature Control

The way in which the process temperature is controlled varies. In a heater, a temperature controller measures the process temperature and signals a fuel valve to either let more or less fuel to the burner. In a heat exchanger the temperature controller could signal a valve to allow more or less of the heating or cooling media to bypass the exchanger.

Flow Control

It is very rare that flow must be controlled in an oil field process. Normally, the control of pressure, level, and temperature is sufficient. Occasionally, it is necessary to assure that flow is split in some controlled manner between two process components in parallel, or perhaps to maintain a certain critical flow through a component. This can become a complicated control problem and must be handled on an individual basis.

BASIC SYSTEM CONFIGURATION

Wellhead and Manifold

The production system begins at the wellhead, which should include at least one choke, unless the well is on artificial lift. Most of the pressure drop between the well flowing tubing pressure (FTP) and the initial separator operating pressure occurs across this choke. The size of the opening in the choke determines the flow rate, because the pressure upstream is determined primarily by the well FTP, and the pressure downstream is determined primarily by the pressure control valve on the first separator in the process. For high-pressure wells it is desirable to have a positive choke in series with an adjustable choke. The positive choke takes over and keeps the production rate within limits should the adjustable choke fail.

On offshore facilities and other high-risk situations, an automatic shutdown valve should be installed on the wellhead. (It is required by federal law in the United States.) In all cases, block valves are needed so that maintenance can be performed on the choke if there is a long flowline.

Whenever flows from two or more wells are commingled in a central facility, it is necessary to install a manifold to allow flow from any one well to be produced into any of the bulk or test production systems.

Separation

Initial Separator Pressure

Because of the multicomponent nature of the produced fluid, the higher the pressure at which the initial separation occurs, the more liquid will be obtained in the separator. This liquid contains some light components that vaporize in the stock tank downstream of the separator. If the pressure for initial separation is too high, too many light compo-

nents will stay in the liquid phase at the separator and be lost to the gas phase at the tank. If the pressure is too low, not as many of these light components will be stabilized into the liquid at the separator and they will be lost to the gas phase.

This phenomenon, which can be calculated using flash equilibrium techniques discussed in Chapter 3, is shown in Figure 2-5. It is important to understand this phenomenon qualitatively. The tendency of any one component in the process stream to flash to the vapor phase depends on its partial pressure. The partial pressure of a component in a vessel is defined as the number of molecules of that component in the vapor space divided by the total number of molecules of all components in the vapor space times the pressure in the vessel. Thus, if the pressure in the vessel is high, the partial pressure for the component will be relatively high and the molecules of that component will tend toward the liquid phase. This is seen by the top line in Figure 2-5. As the separator pressure is increased, the liquid flow rate out of the separator increases.

The problem with this is that many of these molecules are the lighter hydrocarbons (methane, ethane, and propane), which have a strong tendency to flash to the gas state at stock tank conditions (atmospheric pressure). In the stock tank, the presence of these large numbers of molecules creates a low partial pressure for the intermediate range hydrocarbons (butanes, pentane, and heptane) whose flashing tendency at stock tank conditions is very susceptible to small changes in partial pressure. Thus, by keeping the lighter molecules in the feed to the stock tank we manage to capture a small amount of them as liquids, but we lose to the gas phase many more of the intermediate range molecules. That is why beyond some optimum point there is actually a decrease in stock tank liquids by increasing the separator operating pressure.

Stage Separation

Figure 2-5 deals with a simple single-stage process. That is, the fluids are flashed in an initial separator and then the liquids from that separator are flashed again at the stock tank. Traditionally, the stock tank is not normally considered a separate stage of separation, though it most assuredly is.

Figure 2-6 shows a three-stage separation process. The liquid is first flashed at an initial pressure and then flashed at successively lower pressures two times before entering the stock tank.

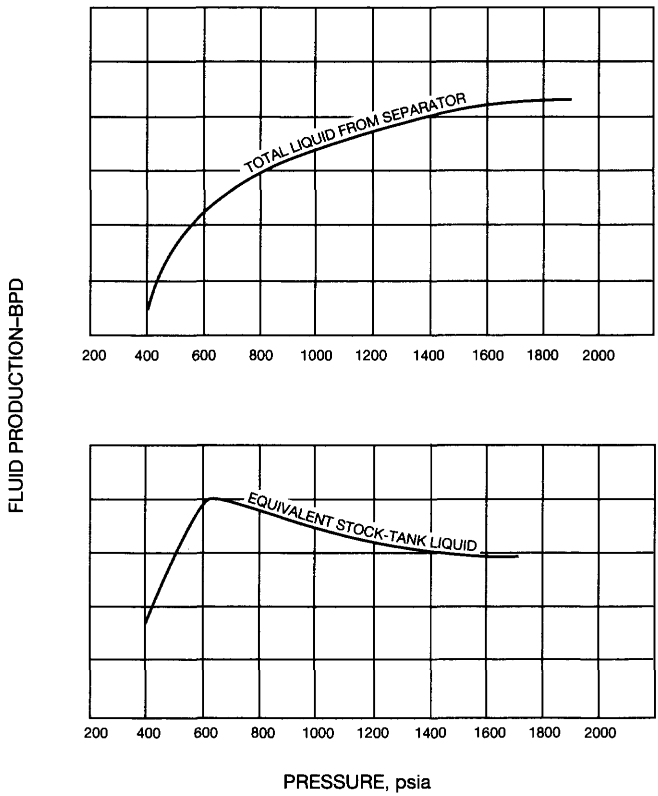


Figure 2-5. Effect of separator pressure on stock-tank liquid recovery.

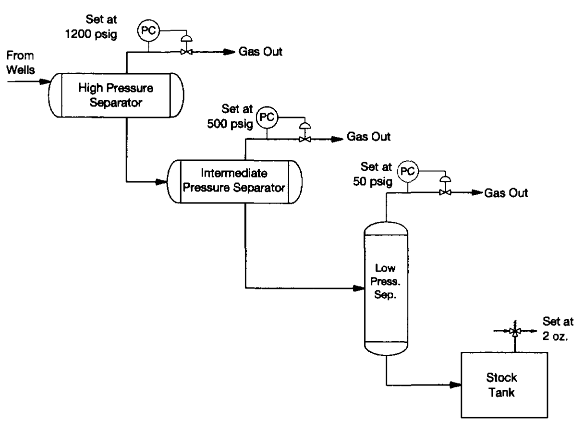


Figure 2-6. Stage separation.

Because of the multicomponent nature of the produced fluid, it can be shown by flash calculations that the more stages of separation after the initial separation the more light components will be stabilized into the liquid phase. This can be understood qualitatively by realizing that in a stage separation process the light hydrocarbon molecules that flash are removed at relatively high pressure, keeping the partial pressure of the intermediate hydrocarbons lower at each stage. As the number of stages approaches infinity, the lighter molecules are removed as soon as they are formed and the partial pressure of the intermediate components is maximized at each stage. The compressor horsepower required is also reduced by stage separation as some of the gas is captured at a higher pressure than would otherwise have occurred. This is demonstrated by the example presented in Table 2-1.

Selection of Stages

As more stages are added to the process there is less and less incremental liquid recovery. The diminishing income for adding a stage must more than offset the cost of the additional separator, piping, controls, space, and compressor complexities. It is clear that for each facility there is an optimum number of stages. In most cases, the optimum number of stages is very difficult to determine as it may be different from well to well and it may change as the wells' flowing pressure declines with time. Table 2-2 is an approximate guide to the number of stages in separation, excluding the stock tank, which field experience indicates is somewhat near optimum. Table 2-2 is meant as a guide and should not replace flash calculations, engineering studies, and engineering judgment.

Table 2-1
Effect of Separation Pressure for a Rich Condensate Stream

Case	Separation Stages psia	Liquid Produced bopd	Compressor Horsepower Required
I	1215, 65	8,400	861
II	1215, 515, 65	8,496	497
III	1215, 515, 190, 65	8,530	399

Table 2-2
Stage Separation Guidelines

Initial Separator Pressure, psig	Number of Stages*
25–125	1
125–300	1–2
300–500	2
500–700	2–3**

* *Does not include stock tank.*
** *At flow rates exceeding 100,000 bopd, more stages may be appropriate.*

Fields with Different Flowing Tubing Pressures

The discussion to this point has focused on a situation where all the wells in a field produce at roughly the same flowing tubing pressure, and stage separation is used to maximize liquid production and minimize compressor horsepower. Often, as in our example flowsheet, stage separation is used because different wells producing to the facility have different flowing tubing pressures. This could be because they are completed in different reservoirs, or are located in the same reservoir but have different water production rates. By using a manifold arrangement and different primary separator operating pressures, there is not only the benefit of stage separation of high-pressure liquids, but also conservation of reservoir energy. High-pressure wells can continue to flow at sales pressure requiring no compression, while those with lower tubing pressures can flow into whichever system minimizes compression.

Separator Operating Pressures

The choice of separator operating pressures in a multistage system is large. For large facilities many options should be investigated before a final choice is made. For facilities handling less than 50,000 bpd, there are practical constraints that help limit the options.

A minimum pressure for the lowest pressure stage would be in the 25 to 50 psig range. This pressure will probably be needed to allow the oil to be dumped to a treater or tank and the water to be dumped to the water treating system. The higher the operating pressure the smaller the compressor needed to compress the flash gas to sales. Compressor horsepower requirements are a function of the absolute discharge pressure divided by the absolute suction pressure. Increasing the low pressure separator pressure from 50 psig to 200 psig may decrease the compression horsepower

required by 33%. However, it may also add backpressure to wells, restricting their flow, and allow more gas to be vented to atmosphere at the tank. Usually, an operating pressure of between 50 and 100 psig is optimum.

As stated before, the operating pressure of the highest pressure separator will be no higher than the sales gas pressure. A possible exception to this could occur where the gas lift pressure is higher than the sales gas pressure. In choosing the operating pressures of the intermediate stages it is useful to remember that the gas from these stages must be compressed. Normally, this will be done in a multistage compressor. For practical reasons, the choice of separator operating pressures should match closely and be slightly greater than the compressor interstage pressures. The most efficient compressor sizing will be with a constant compressor ratio per stage. Therefore, an approximation of the intermediate separator operating pressures can be derived from:

$$R = \left[\frac{P_d}{P_s}\right]^{1/n} \tag{2-1}$$

where R = ratio per stage
P_d = discharge pressure, psia
P_s = suction pressure, psia
n = number of stages

Once a final compressor selection is made, these approximate pressures will be changed slightly to fit the actual compressor configuration.

In order to minimize interstage temperatures the maximum ratio per stage will normally be in the range of 3.6 to 4.0. That means that most production facilities will have either two- or three-stage compressors. A two-stage compressor only allows for one possible intermediate separator operating pressure. A three-stage allows for either one operating at second- or third-stage suction pressure, or two intermediate separators each operating at one of the two compressor intermediate suction pressures. Of course, in very large facilities it would be possible to install a separate compressor for each separator and operate as many intermediate pressure separators as is deemed economical.

Two Phase vs. Three Phase Separators

In our example process the high- and intermediate-stage separators are two-phase, while the low-pressure separator is three-phase. This is called

a "free water knockout" (FWKO) because it is designed to separate the free water from the oil and emulsion, as well as separate gas from liquid. The choice depends on the expected flowing characteristics of the wells. If large amounts of water are expected with the high-pressure wells, it is possible that the size of the other separators could be reduced if the high-pressure separator was three-phase. This would not normally be the case for a facility such as that shown in Figure 2-1 where individual wells are expected to flow at different FTPs. In some instances, where all wells are expected to have similar FTPs at all times, it may be advantageous to remove the free water early in the separation scheme.

Process Flowsheet

Figure 2-7 is an enlargement of the FWKO of Figure 2-1 to show the amount of detail that would be expected on a process flowsheet. A flash calculation is needed to determine the amount of gas and liquid that each separator must handle.

In the example process of Figure 2-1, the treater is not considered a separate stage of separation as it operates very close to the FWKO pressure, which is the last stage. Very little gas will flash between the two vessels. In most instances, this gas will be used for fuel or vented and not

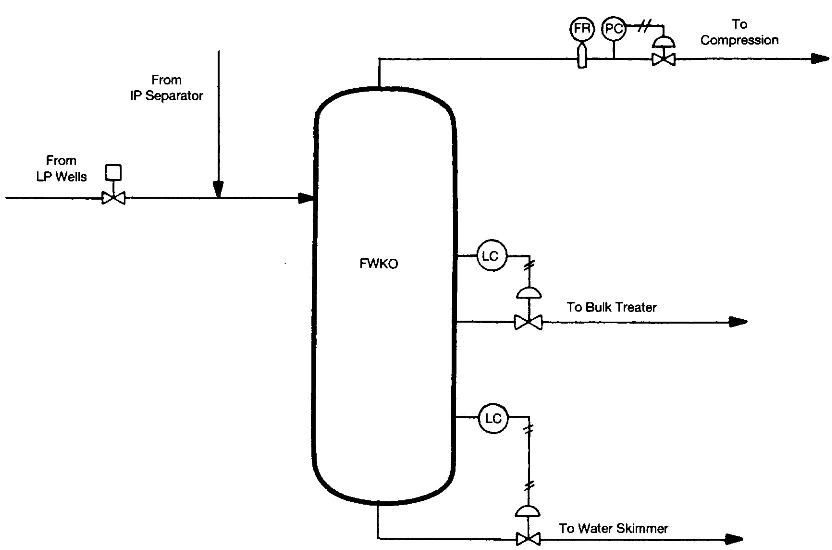


Figure 2-7. Free water knockout.

compressed for sales, although a small compressor could be added to boost this gas to the main compressor suction pressure.

Oil Treating

Most oil treating on offshore facilities is done in vertical or horizontal treaters, such as those described in Chapter 6. Figure 2-8 is an enlargement of the oil treater in Figure 2-1. In this case, a gas blanket is provided to assure that there is always enough pressure in the treater so the water will flow to water treating.

At onshore locations the oil may be treated in a big "gunbarrel" (or settling) tank, as shown in Figure 2-9. All tanks should have a pressure/vacuum valve with flame arrestor and gas blanket to keep a positive pressure on the system and exclude oxygen. This helps to prevent corrosion, eliminate a potential safety hazard, and conserve some of the hydrocarbon vapors. Figure 2-10 shows a typical pressure/vacuum valve. A pressure in the tank lifts a weighted disk or pallet, which allows the gas to escape. If there is a vacuum in the tank because the gas blanket failed to maintain a slight positive pressure, the greater ambient pressure lifts another disk, which allows air to enter. Although we wish to exclude air, it is preferable to allow a small controlled volume into the tank rather than allow the tank to collapse. The savings associated with keeping a positive pressure on the tank is demonstrated by Table 2-3.

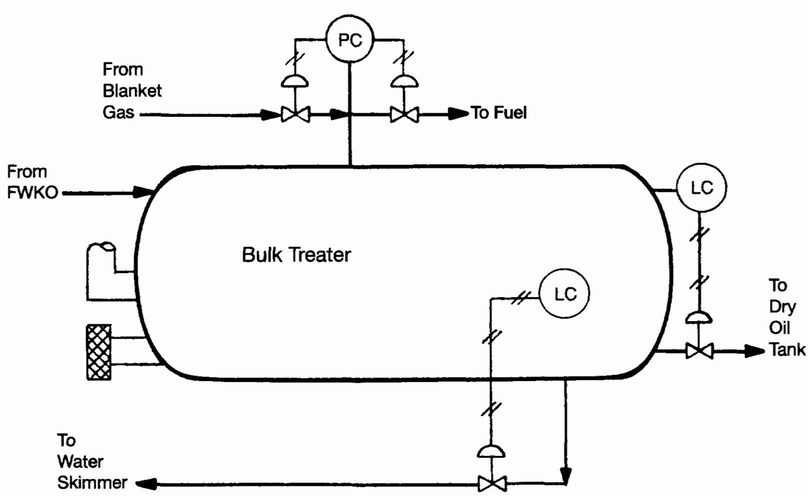


Figure 2-8. Bulk treater.

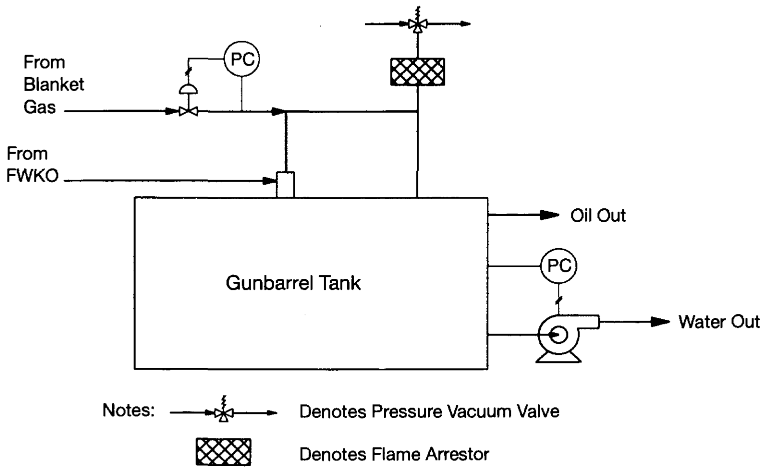


Figure 2-9. "Gunbarrel."

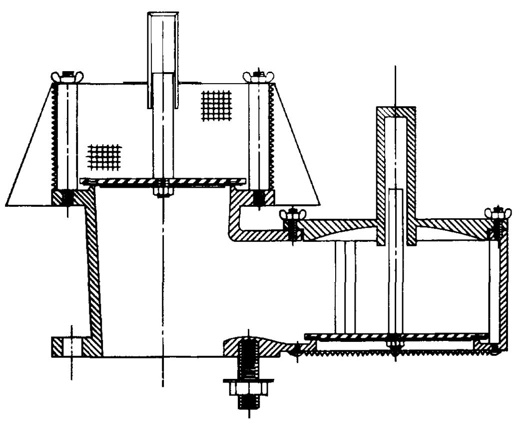


Figure 2-10. Typical pressure/vacuum valve (*courtesy of Groth Equipment Corp.*).

Table 2-3
Tank Breathing Loss

Nominal Capacity bbl	Breathing Loss Open Vent bbl/yr	Pressure Valve bbl/yr	Barrels Saved
5,000	235	154	81
10,000	441	297	144
20,000	625	570	255
55,000	2,000	1,382	618

Figure 2-11 shows a typical flame arrestor. The tubes in the device keep a vent flame from traveling back into the tank. Flame arrestors have a tendency to plug with paraffin and thus must be installed where they can be inspected and maintained. Since they can plug, a separate relieving device (most often a gauge hatch set to open a few ounces above the normal relieving device) must always be installed.

The oil is skimmed off the surface of the gun barrel and the water exits from the bottom either through a water leg or an interface controller and dump valve. It must be pointed out that since the volume of the liquid is fixed by the oil outlet, gun barrels cannot be used as surge tanks.

Flow from the treater or gun barrel goes to a surge tank from which it either flows into a barge or truck or is pumped into a pipeline.

Lease Automatic Custody Transfer (LACT)

In large facilities oil is typically sold through a LACT unit, which is designed to meet API Standards and whatever additional measuring and sampling standards are required by the crude purchaser. The value

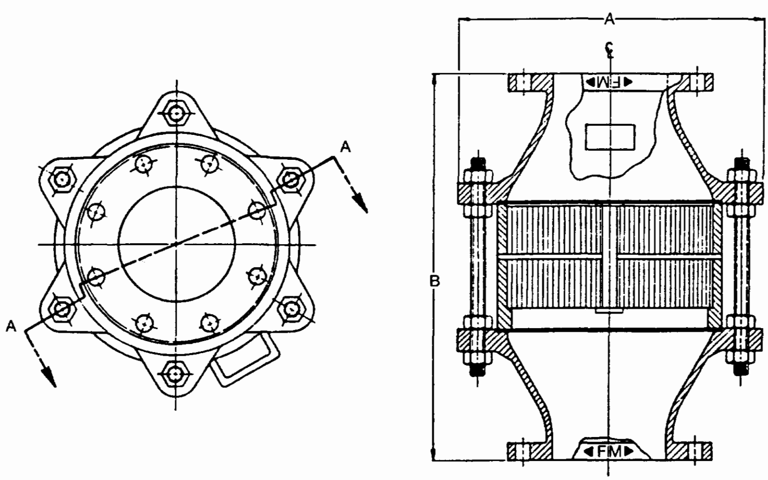


Figure 2-11. Typical flame arrestor (*courtesy of Groth Equipment Corp.*).

received for the crude will typically depend on its gravity, BS+W content, and volume. Therefore, the LACT unit must not only measure the volume accurately, but must continuously monitor the BS+W content and take a sufficiently representative sample so that the gravity and BS+W can be measured.

Figure 2-12 shows schematically the elements of a typical LACT unit. The crude first flows through a strainer/gas eliminator to protect the meter and to assure that there is no gas in the liquid. An automatic BS+W probe is mounted in a vertical run. When BS+W exceeds the sales contract quality this probe automatically actuates the diverter valve, which blocks the liquid from going further in the LACT unit and sends it back to the process for further treating. Some sales contracts allow for the BS+W probe to merely sound a warning so that the operators can manually take corrective action. The BS+W probe must be mounted in a vertical run if it is to get a true reading of the average quality of the stream.

Downstream of the diverter a sampler in a vertical run takes a calibrated sample that is proportional to the flow and delivers it to a sample container. The sampler receives a signal from the meter to assure that the sample size is always proportional to flow even if the flow varies. The sample container has a mixing pump so that the liquid in the container can be mixed and made homogeneous prior to taking a sample of this fluid. It is this small sample that will be used to convert the meter reading for BS+W and gravity.

The liquid then flows through a positive displacement meter. Most sales contracts require the meter to be proven at least once a month and a new meter factor calculated. On large installations a meter prover such as that shown in Figure 2-12 is included as a permanent part of the LACT skid or is brought to the location when a meter must be proven. The meter prover contains a known volume between two detector switches. This known volume has been measured in the factory to ±0.02% when measured against a tank that has been calibrated by the National Bureau of Standards. A spheroid pig moves back and forth between the detectors as the four-way valve is automatically switched. The volume recorded by the meter during the time the pig moves between detectors for a set number of traverses of the prover is recorded electrically and compared to the known volume of the meter prover.

On smaller installations, a master meter that has been calibrated using a prover may be brought to the location to run in series with the meter to be proven. In many onshore locations a truck-mounted meter prover is

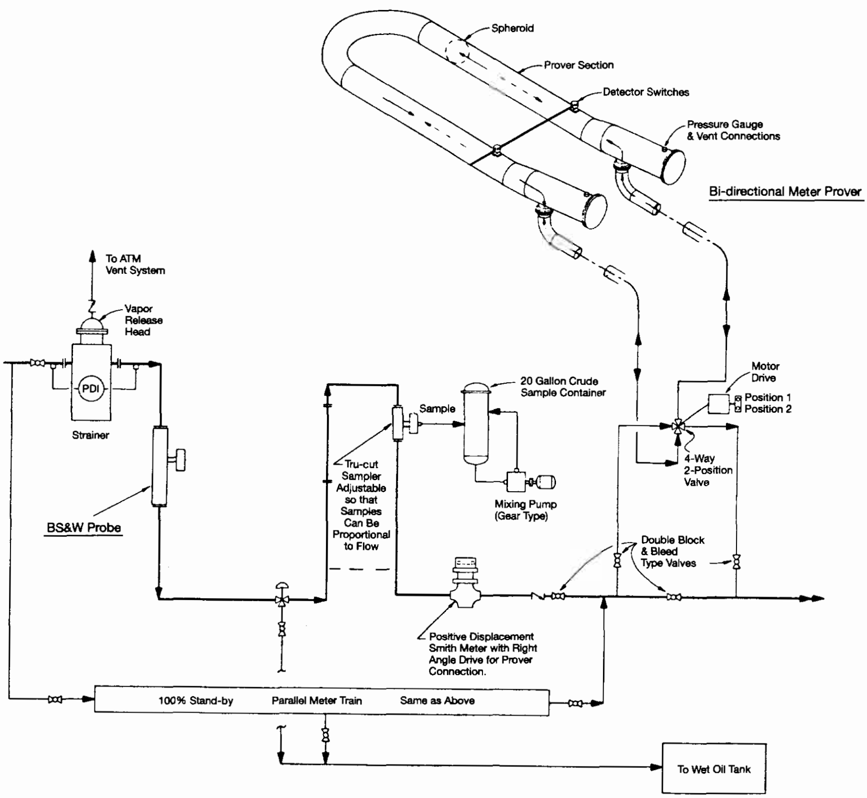


Figure 2-12. Typical LACT unit schematic.

used. The sales meter must have a proven repeatability of ±0.02% when calibrated against a master meter or ±0.05% when calibrated against a tank or meter prover.

Pumps

Pumps are normally needed to move oil through the LACT unit and deliver it at pressure to a pipeline downstream of the unit. Pumps are sometimes used in water treating and disposal processes. In addition, many small pumps may be required for pumping skimmed oil to higher pressure vessels for treating, glycol heat medium and cooling water service, firefighting, etc. The various types of pumps and their uses are further explained in Chapter 10.

Water Treating

Chapter 7 describes choosing a process for this subsystem, including vessel and open drains. Figure 2-13 shows an enlargement of the water treating system for the example.

Compressors

Figure 2-14 shows the configuration of the typical three-stage reciprocating compressor in our example flowsheet. Gas from the FWKO enters the first-stage suction scrubber. Any liquids that may have come through the line are separated at this point and the gas flows to the first stage. Compression heats the gas, so there is a cooler after each compression stage. At the higher pressure more liquids may separate, so the gas enters another scrubber before being compressed and cooled again.

In the example, gas from the intermediate pressure separator can be routed to either the second-stage or third-stage suction pressure, as conditions in the field change.

Gulf of Mexico accident records indicate that compressors are the single most hazardous piece of equipment in the process. The compressor is equipped with an automatic suction shut-in valve on each inlet and a discharge shut-in valve so that when the unit shuts down, or when an abnormal condition is detected, the shut-in valves actuate to isolate the unit from any new sources of gas. Many operators prefer, and in some cases regulations require, that an automatic blowdown valve also be installed so that as well as isolating the unit, all the gas contained within the unit is vented safely at a remote location.

Compressors in oil field service should be equipped with a recycle valve and a vent valve, such as shown in Figure 2-14. Compressor operating conditions are typically not well known when the compressor is installed, and even if they were, they are liable to change greatly as wells come on and off production. The recycle valve allows the compressor to be run at low throughput rates by keeping the compressor loaded with its minimum required throughput. In a reciprocating compressor this is done by maintaining a minimum pressure on the suction. In a centrifugal compressor this is done by a more complex surge control system.

The vent valve allows production to continue when the compressor shuts down. Many times a compressor will only be down for a short time

(text continued on page 46)

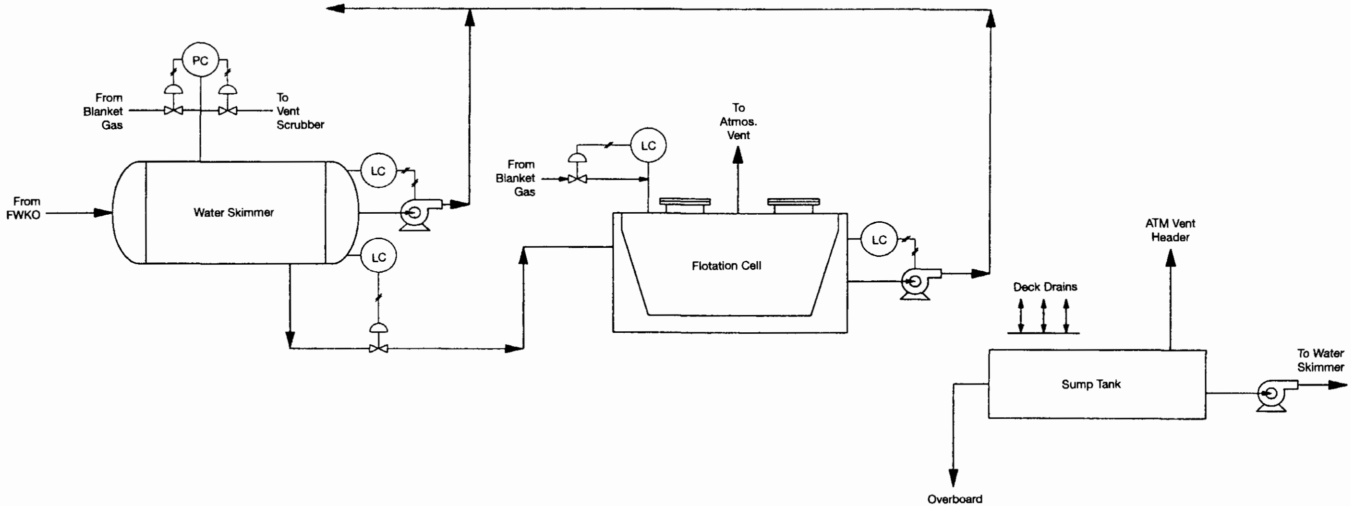


Figure 2-13. Water treating system.

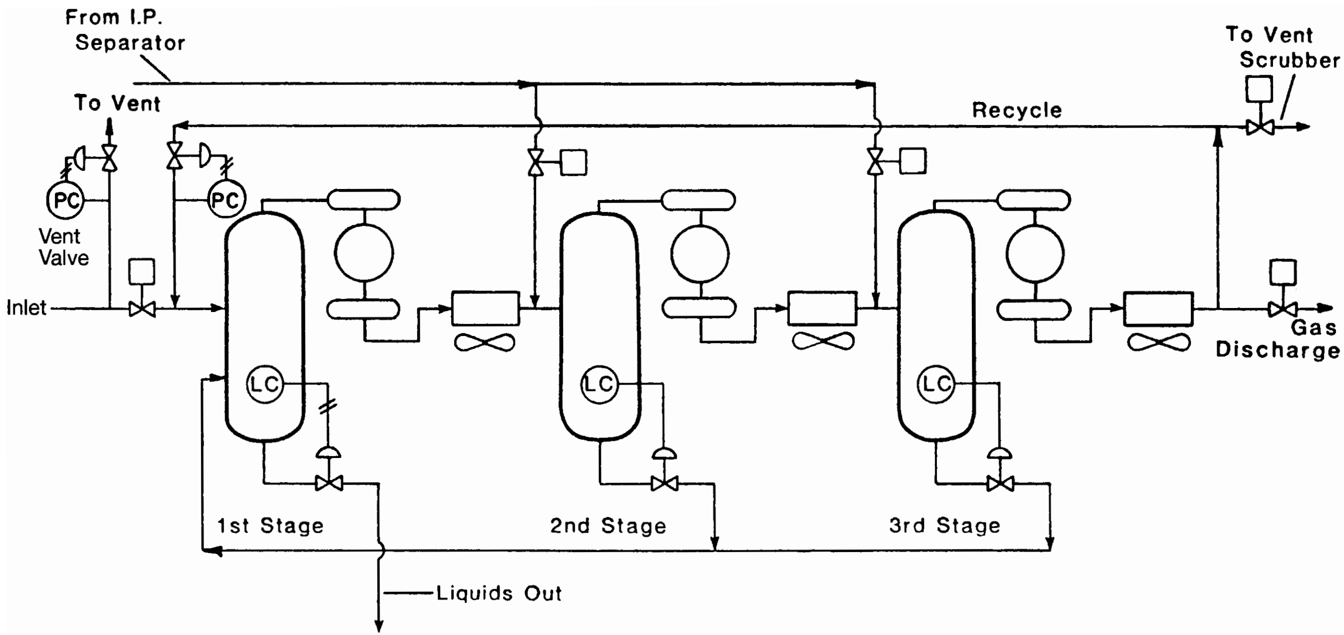


Figure 2-14. Three-stage compressor.

(text continued from page 43)

and it is better to vent the gas rather than automatically shut in production. The vent valve also allows the compressor to operate when there is too much gas to the inlet. Under such conditions the pressure will rise to a point that could overload the rods on a reciprocating compressor.

The two basic types of compressors used in production facilities are reciprocating and centrifugal. Reciprocating compressors compress the gas with a piston moving linearly in a cylinder. Because of this, the flow is not steady, and care must be taken to control vibrations. Centrifugal compressors use high-speed rotating wheels to create a gas velocity that is converted into pressure by stators.

Reciprocating compressors are particularly attractive for low horsepower (< 2,000 hp), high-ratio applications, although they are available in sizes up to approximately 10,000 hp. They have higher fuel efficiencies than centrifugals, and much higher turndown capabilities.

Centrifugal compressors are particularly well suited for high horsepower (>4,000 hp) or for low ratio (<2.5) in the 1,000 hp and greater sizes. They are less expensive, take up less space, weigh less, and tend to have higher availability and lower maintenance costs than recips. Their overall fuel efficiency can be increased if use is made of the high temperature exhaust heat in the process.

Gas Dehydrators

Removing most of the water vapor from the gas is required by most gas sales contracts, because it prevents hydrates from forming when the gas is cooled in the transmission and distribution systems and prevents water vapor from condensing and creating a corrosion problem. Dehydration also increases line capacity marginally.

Most sales contracts in the southern United States call for reducing the water content in the gas to less than 7 lb/MMscf. In colder climates, sales requirements of 3 to 5 lb/MMscf are common. The following methods can be used for drying the gas:

1. Cool to the hydrate formation level and separate the water that forms. This can only be done where high water contents (±30 lb/MMscfd) are acceptable.
2. Use a Low-temperature Exchange (LTX) unit designed to melt the hydrates as they are formed. Figure 2-15 shows the process. LTX

units require inlet pressures greater than 2,500 psi to work effectively. Although they were common in the past, they are not normally used because of their tendency to freeze and their inability to operate at lower inlet pressure as the well FTP declines.

3. Contact the gas with a solid bed of $CaCl_2$. The $CaCl_2$ will reduce the moisture to low levels, but it cannot be regenerated and is very corrosive.
4. Use a solid desiccant, such as activated alumina, silica gel or molecular sieve, which can be regenerated. These are relatively expensive units, but they can get the moisture content to very low levels. Therefore, they tend to be used on the inlets to low temperature gas processing plants, but are not common in production facilities.

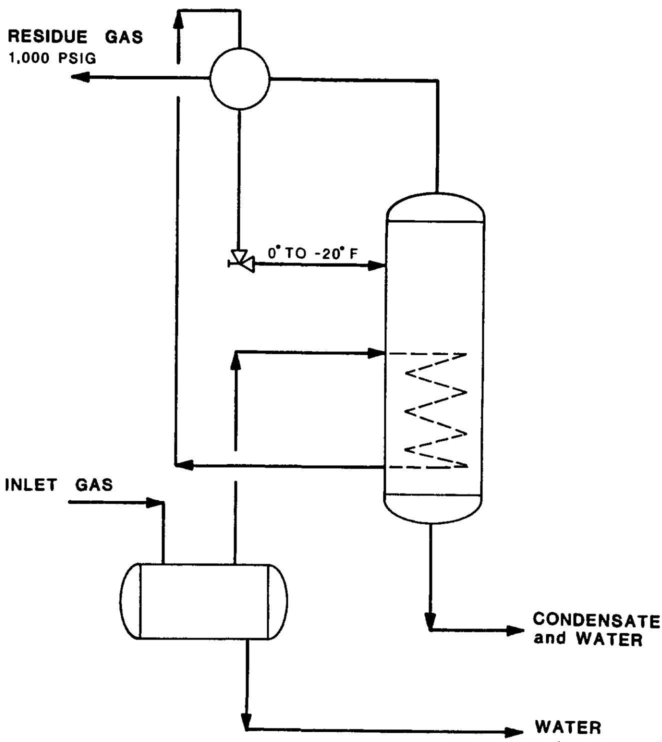


Figure 2-15. Low-temperature exchange unit.

5. Use a liquid desiccant, such as methanol or ethylene glycol, which cannot be regenerated. These are relatively inexpensive. Extensive use is made of methanol to lower the hydrate temperature of gas well flowlines to keep hydrates from freezing the choke.
6. Use a glycol liquid desiccant, which can be regenerated. This is the most common type of gas dehydration system and is the one shown on the example process flowsheet.

Figure 2-16 shows how a typical bubble-cap glycol contact tower works. Wet gas enters the base of the tower and flows upward through the bubble caps. Dry glycol enters the top of the tower, and because of the downcomer wier on the edge of each tray, flows across the tray and down to the next. There are typically six to eight trays in most applications. The bubble caps assure that the upward flowing gas is dispersed into small bubbles to maximize its contact area with the glycol.

Before entering the contactor the dry glycol is cooled by the outlet gas to minimize vapor losses when it enters the tower. The wet glycol leaves from the base of the tower and flows to the reconcentrator (reboiler) by way of heat exchangers, a gas separator, and filters, as shown in Figure 2-17. In the reboiler the glycol is heated to a sufficiently high temperature to drive off the water as steam. The dry glycol is then pumped back to the contact tower.

Most glycol dehydrators use triethylene glycol, which can be heated to 340°F to 400°F in the reconcentrator and work with gas temperatures up to 120°F. Tetraethylene glycol is more expensive but it can handle hotter gas without high losses and can be heated in the reconcentrator to 400°F to 430°F.

WELL TESTING

It is necessary to keep track of the gas, oil, and water production from each well to be able to manage the reserves properly, evaluate where further reserve potential may be found, and diagnose well problems as quickly as possible. Proper allocation of income also requires knowledge of daily production rates as the royalty or working interest ownership may be different for each well.

In simple facilities that contain only a few wells, it is attractive to route each well to its own separator and/or treater and measure its gas, oil, and water production on a continuous basis. In facilities that handle

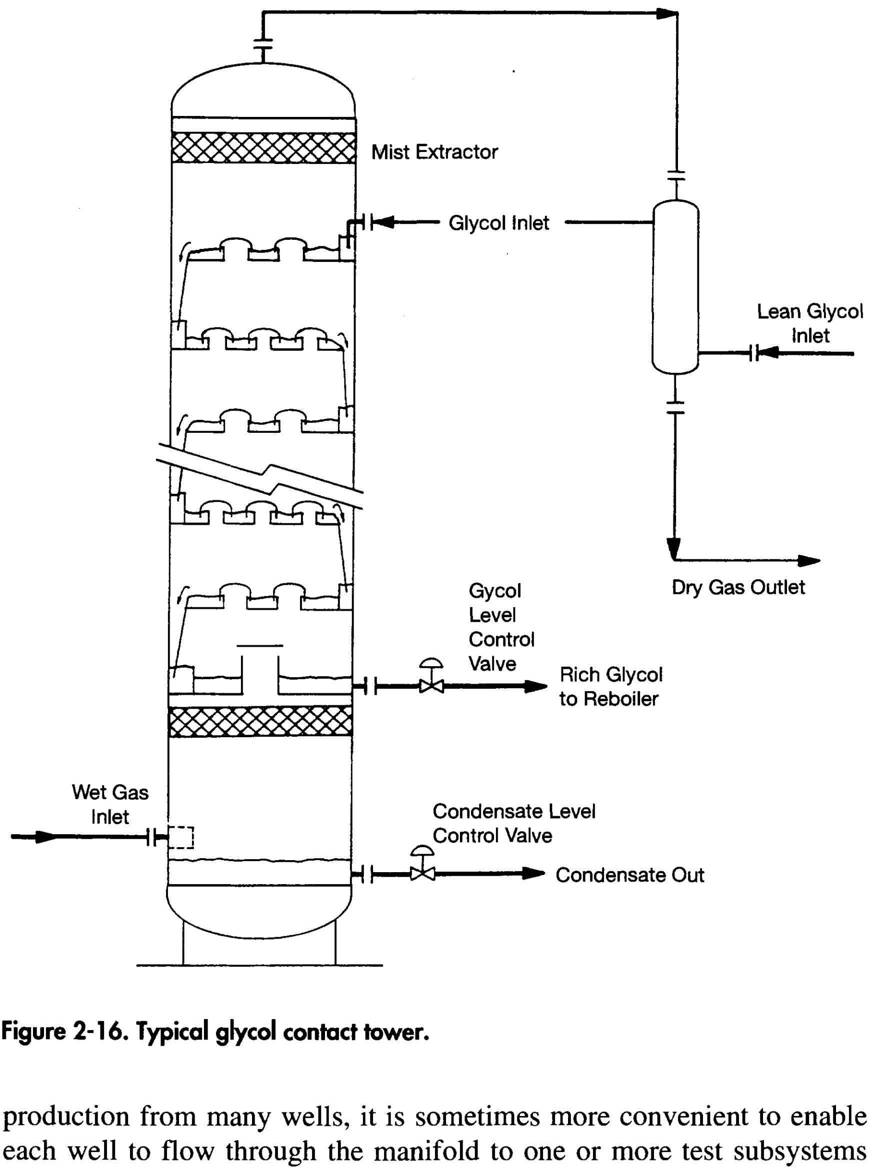


Figure 2-16. Typical glycol contact tower.

production from many wells, it is sometimes more convenient to enable each well to flow through the manifold to one or more test subsystems on a periodic basis. Total production from the facility is then allocated back to the individual wells on the basis of these well tests.

The frequency with which wells must be tested and the length of the test depends upon well properties, legal requirements, requirements for special studies, etc. Most oil wells should be tested at least twice a month for four to twelve hours. Gas wells should be tested at least once a

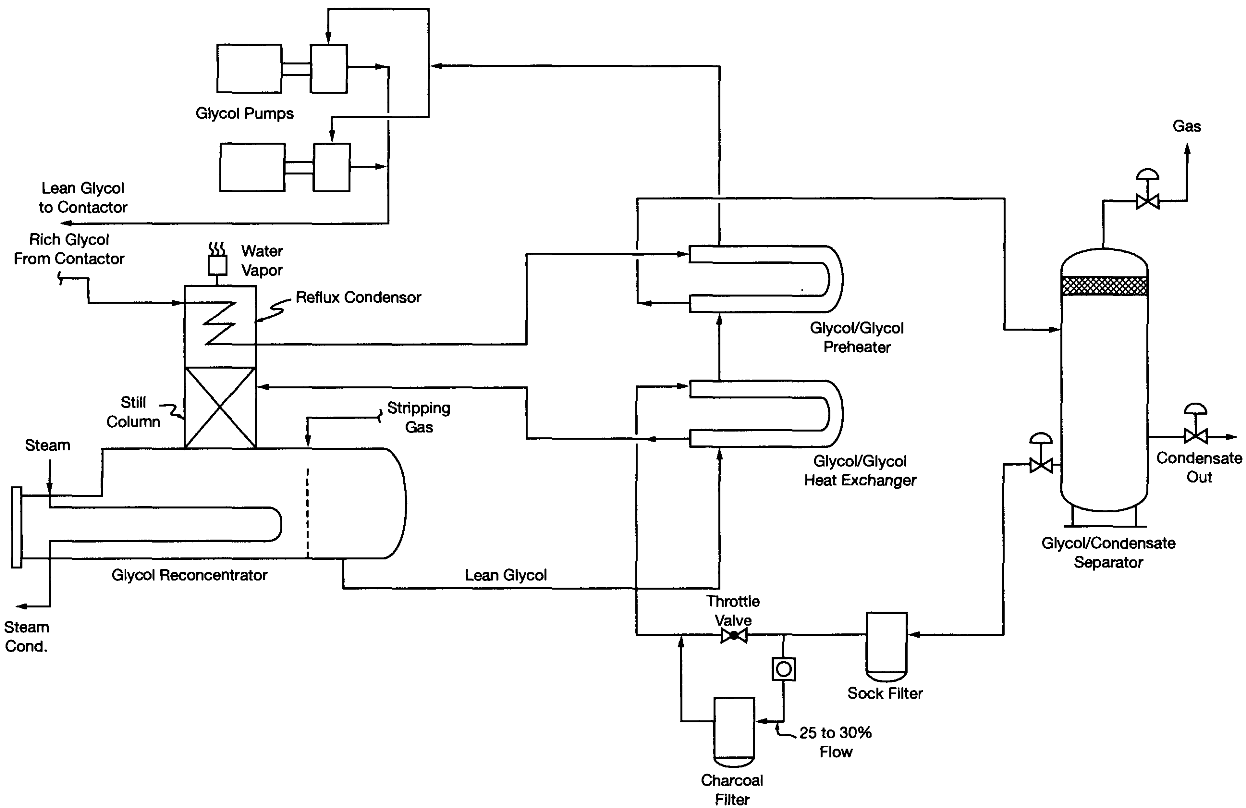


Figure 2-17. Typical glycol reconcentrator.

month. Due to the need to put troublesome wells on long-term tests, the need to repeat tests whose results might be suspect, and the need to test several wells whenever there is an unexpected change in total production, one test system can handle approximately 20 oil wells.

In order to obtain a valid test, the test system should operate at the same pressure as the system to which the well normally flows. That is, if a well normally flows to a high-pressure separator the first vessel in the test system should operate at that pressure. If other wells normally flow to an intermediate- or low-pressure separator, the first vessel in the test system must be able to operate at that pressure as well. Thus, in our example facility, Figure 2-1, we must either install separate high-, intermediate- and low-pressure test systems, or we must arrange the gas backpressure valves on the first vessel in the test system so that the vessel can operate at any of the three pressures by just switching a valve.

A test system can be made up of any of the components we have discussed (e.g., separators, FWKOs, treaters) arranged in any combination that makes sense to obtain the required data. A three-phase separator could be used where oil/water emulsions are not considered severe. The amount of oil in the water outlet is insignificant and can be neglected. The water in the oil outlet can be determined from a net oil computer, which automatically corrects for the water, or by taking a sample and measuring its oil content. This would be particularly well suited for gas wells.

A vertical treater could be used where it was considered necessary to heat the emulsion in order to measure its water content. Standard treaters are low-pressure vessels with limited gas and free water capacity. For this reason they would tend to be used on low-pressure oil wells. If it is desirable to use the treater on a higher pressure oil well, this could be done by including a separator upstream of and in series with the treater. If a great deal of free water is expected, the treater could be designed with a large FWKO section or a three-phase separator could be installed upstream.

Some facilities use a high-pressure three-phase separator for the high- and intermediate-pressure wells that do not make much water and a treater for the low-pressure wells. Figure 2-18 shows an enlargement of the well test system for the example.

GAS LIFT

We must comment a bit about gas lift systems because they are in widespread use and have a significant impact on the facility process. Fig-

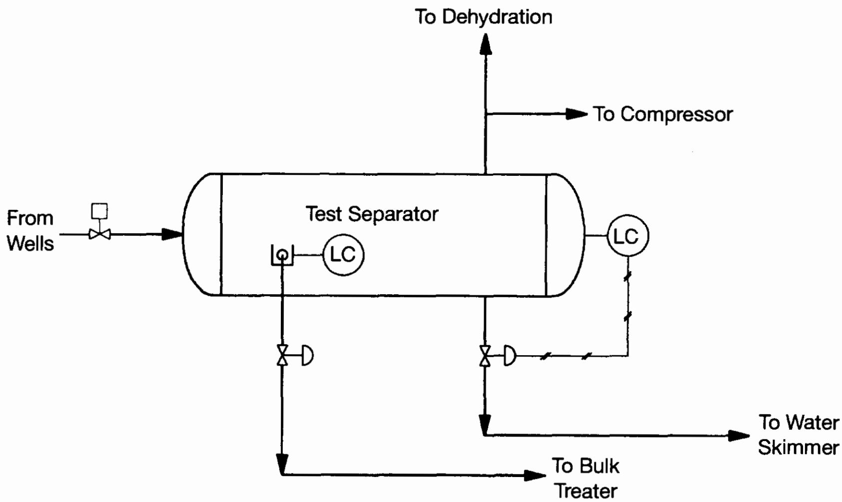


Figure 2-18. Well test system.

ure 2-19 is a diagram of a gas lift system from the facility engineer's perspective. High-pressure gas is injected into the well to lighten the column of fluid and allow the reservoir pressure to force the fluid to the surface. The gas that is injected is produced with the reservoir fluid into the low-pressure system. Therefore, the low-pressure separator must have sufficient gas separation capacity to handle gas lift as well as formation gas.

If gas lift is to be used, it is even more important from a production standpoint that the low-pressure separator be operated at the lowest prac-

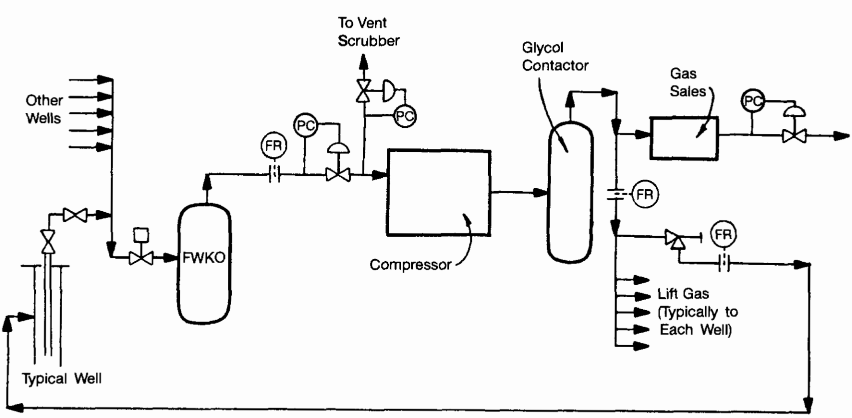
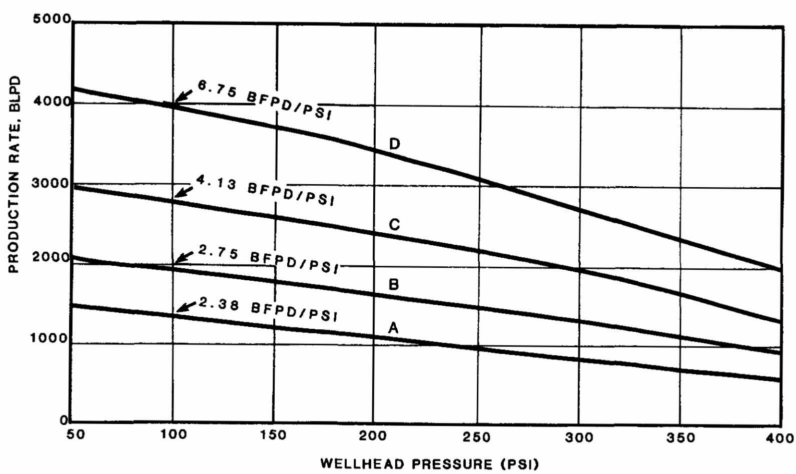


Figure 2-19. Gas lift system.

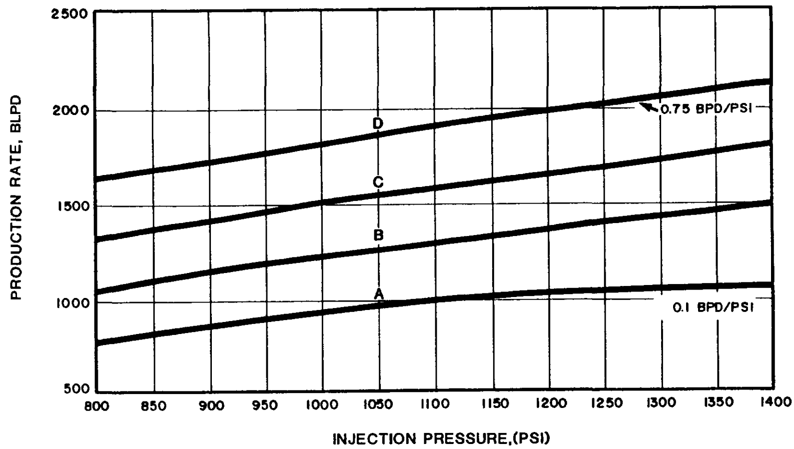
tical pressure. Figure 2-20 shows the effects of wellhead backpressure for a specific set of wells. It can be seen that a one psi change in well backpressure will cause between 2 and 6 BFPD change in well deliverability.

The higher the injected gas pressure into the casing the deeper the last gas lift valve can be set. As shown in Figure 2-21, for a typical well the higher the design injection the higher the flowrate. Most gas sales contracts are in the 1,000 to 1,200 psi range. Therefore, the process must be designed to deliver the sales gas at this pressure. As seen from Figure 2-21 at about this range a rather large change in gas injection pressure is necessary for a small change in well deliverability. In the range of pressures under consideration (approximately 65-psia suction, 1,215-psia discharge) a 1-psi change in suction pressure (i.e., low-pressure separation operating pressure) is equivalent to a 19-psi change in discharge pressure (i.e., gas lift injection pressure) as it affects compressor ratio and thus compressor horsepower requirements. A comparison of Figures 2-20 and 2-21 shows that a 1-psi lowering of suction pressure in this typical case is more beneficial than a 19-psi increase in discharge pressure for the wells with a low productivity index (PI) but not as beneficial for the high PI wells. The productivity index is the increase in fluid flow into the bottom of the well (in barrels per day) for a 1-psi drawdown in bottomhole pressure.



Note: These curves are for a specific set of tubing size, casing pressure, and fluid out.

Figure 2-20. Effect of wellhead backpressure on total fluid production rate for a specific set of wells.



Note: These curves are for a specific set of tubing size, casing pressure, and fluid out.

Figure 2-21. Effect of gas lift injection pressure on total fluid production rate for a specific set of wells.

Figure 2-22 shows the effect of gas injection rate. As more gas is injected, the weight of fluid in the tubing decreases and the bottomhole flowing pressure decreases. This is balanced by the friction drop in the tubing. As more gas lift gas is injected, the friction drop of the mixture returning to the surface increases exponentially. At some point the friction drop effect is greater than the effect of lowering fluid column weight. At this point, injecting greater volumes of gas lift gas causes the bottomhole pressure to increase and thus the production rate to decrease.

Each gas lift system must be evaluated for its best combination of injection rate, separator pressure, and injection pressure, taking into account process restraints (e.g., need to move the liquid through the process) and the sales gas pressure. In the vast majority of cases, a low-pressure separator pressure of about 50 psig and a gas lift injection pressure of 1,000 to 1,400 psig will prove to be near optimum.

OFFSHORE PLATFORM CONSIDERATIONS

Overview

An increasing amount of the world's oil and gas comes from offshore fields. This trend will accelerate as onshore fields are depleted. A grow-

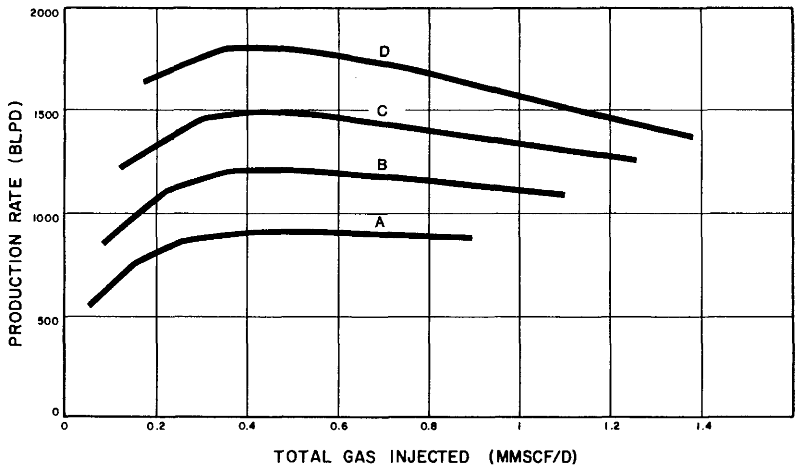


Figure 2-22. Effect of gas lift injection rate on total fluid production rate for a specific set of wells.

ing amount of engineering effort is being spent on offshore facility designs. Thus, it is appropriate that this section describe platforms that accommodate simultaneous drilling and production operations.

Modular Construction

Modules are large boxes of equipment installed in place and weighing from 300 to 2,000 tons each. Modules are constructed, piped, wired and tested in shipyards or in fabrication yards, then transported on barges and set on the platform, where the interconnections are made (Figure 2-23). Modular construction is used to reduce the amount of work and the number of people required for installation and start-up.

Equipment Arrangement

The equipment arrangement plan shows the layout of all major equipment. As shown in Figure 2-24, the right-hand module contains the flare drums, water skimmer tank and some storage vessels. In addition, it provides support for the flare boom.

The adjacent wellhead module consists of a drilling template with conductors through which the wells will be drilled. The third unit from the right contains the process module, which houses the separators and other processing equipment. The fourth and fifth modules from the right con-

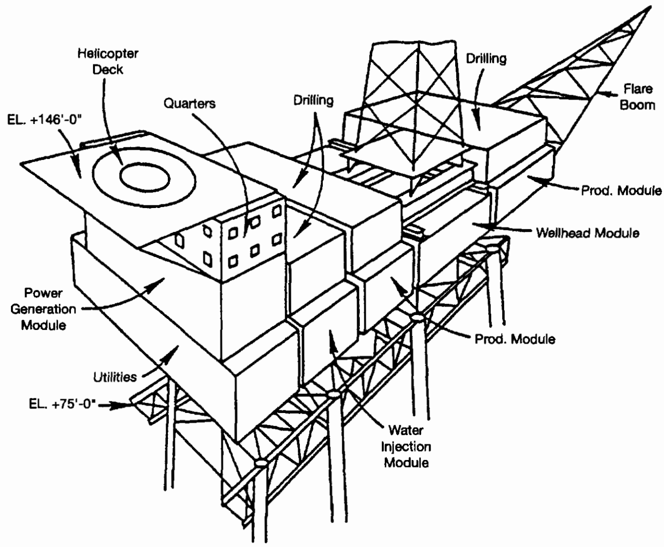


Figure 2-23. Schematic of a large offshore platform, illustrating the concept of modularization.

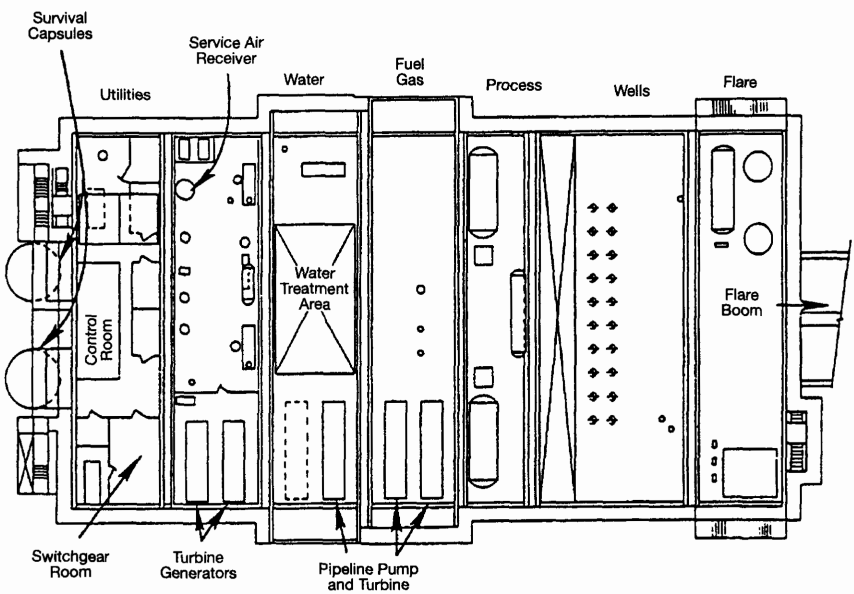


Figure 2-24. Equipment arrangement plan of a typical offshore platform illustrating the layout of the lower deck.

tain turbine-driven pumps, fuel gas scrubbers, and the produced-water treating area. The last two modules house utilities such as power generators, air compressors, potable water makers, a control room, and switchgear and battery rooms. The living quarters are located over the last module.

Figure 2-25 shows an elevation of a platform in which the equipment arrangement is essentially the same.

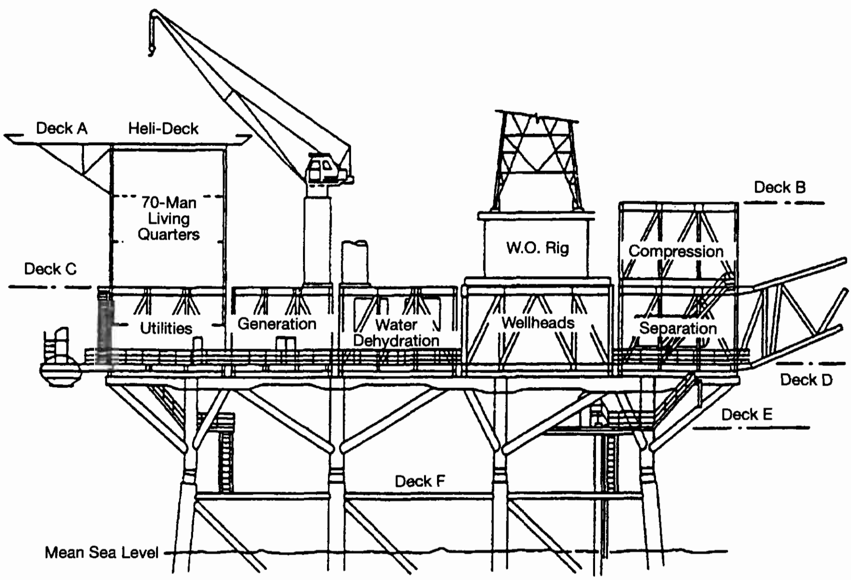


Figure 2-25. Typical elevation view of an offshore platform showing the relationships among the major equipment modules.

CHAPTER

3

*Fluid Properties**

INTRODUCTION

Before describing the equipment used in the process and design techniques for sizing and specifying that equipment, it is necessary to define some basic fluid properties. We will also discuss some of the common calculation procedures, conversions, and operations used to describe the fluids encountered in the process.

BASIC PRINCIPLES

Specific Gravity and Density

Specific gravity of a liquid is the ratio of the density of the liquid at 60°F to the density of pure water. API gravity is related to the specific gravity by the following equation:

$$°API = \frac{141.5}{S.G.} - 131.5 \qquad (3-1)$$

where S.G. = specific gravity of a liquid (water = 1)

*Reviewed for the 1998 edition by Michael Hale of Paragon Engineering Services, Inc.

The specific gravity of a gas is the ratio of the density of the gas to the density of air at standard conditions of temperature and pressure. It may be related to the molecular weight by the following equation:

$$S = \frac{MW}{29} \tag{3-2}$$

where S = specific gravity of a gas (air = 1)
MW = molecular weight

In most calculations the specific gravity of the gas is always referred to in terms of standard conditions of temperature and pressure and therefore is always given by Equation 3-2 once the molecular weight of the gas is known. The density of a gas at any condition of temperature and pressure can be determined by remembering that the density of air at standard conditions of temperature and pressure (60°F and 14.7 psia) is 0.0764 lb/ft^3.

The density of gas is thus given as:

$$\rho_g = 2.70 \frac{SP}{TZ} \tag{3-3}$$

$$\rho_g = 0.093 \frac{(MW)P}{TZ} \tag{3-4}$$

where ρ_g = density of gas, lb/ft^3
S = specific gravity of gas (air = 1)
P = pressure, psia
T = temperature, °R
Z = gas compressibility factor
MW = gas molecular weight

Derivation of Equations 3-3 and 3-4

Density of gas at standard conditions of temperature and pressure is by definition:

$$\rho_{std} = 0.0764\ S$$

The volume of a pound of gas is given by the specific volume as:

$$\overline{V} = \frac{1}{\rho}$$

The equation of state for a gas is given for engineering calculations as:

$$\frac{PV}{ZT} = \text{constant}$$

For any gas, comparing its equation of state at standard conditions to that at actual conditions:

$$\frac{P_{std}\,\overline{V}_{std}}{T_{std}\,Z_{std}} = \frac{P\overline{V}}{TZ}$$

$$\frac{(14.7)}{(520)(1.0)(0.0764)S} = \frac{P}{\rho TZ}$$

$$\rho = 2.70\,\frac{SP}{TZ}$$

$$S = \frac{MW}{29}$$

$$\rho = 0.093\,\frac{(MW)P}{TZ}$$

The compressibility factor for a natural gas can be approximated from Figures 3-1 through 3-6, which are from the *Engineering Data Book* of the Gas Processor Suppliers Association.

In most calculations the specific gravity of liquids is normally referenced to actual temperature and pressure conditions. Figures 3-7 and 3-8 can be used to approximate how the specific gravity of a liquid decreases with increasing temperature, assuming no phase changes. In most practical pressure drop calculations associated with production facilities, the difference in specific gravity caused by pressure changes will not be severe enough to be considered if there are no phase changes.

For hydrocarbons, which undergo significant phase changes, Figure 3-8 can be used as an approximation of the specific gravity at a given pressure and temperature, once the API gravity of the liquid is known.

It should be pointed out that both Figures 3-7 and 3-8 are approximations only for the liquid component. Where precise calculation is required for a hydrocarbon, it is necessary to consider the gas that is liberated with decreasing pressure and increasing temperature. Thus, if a

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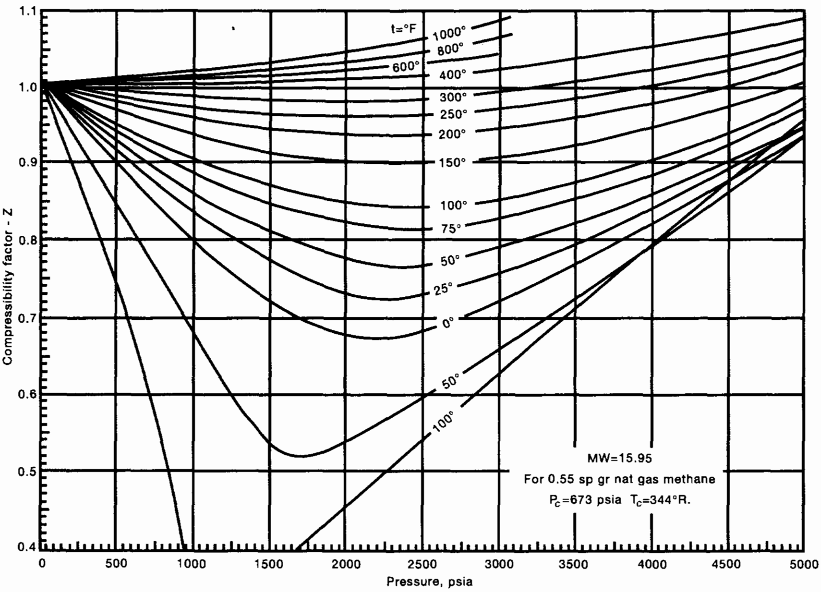

Figure 3-1. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

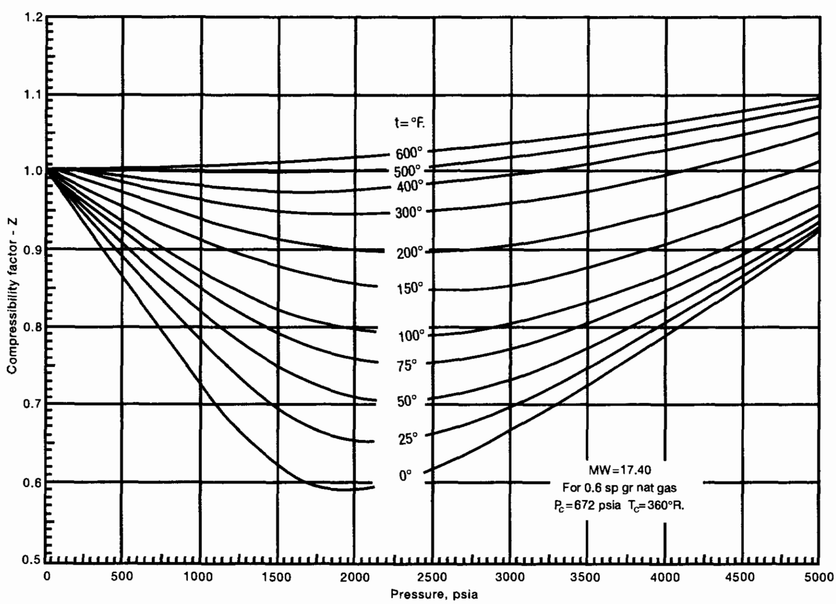

Figure 3-2. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

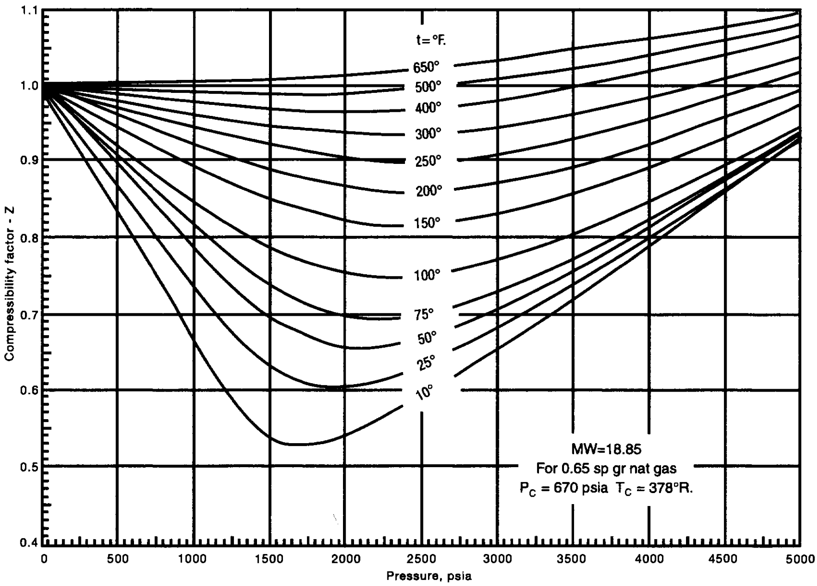

Figure 3-3. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

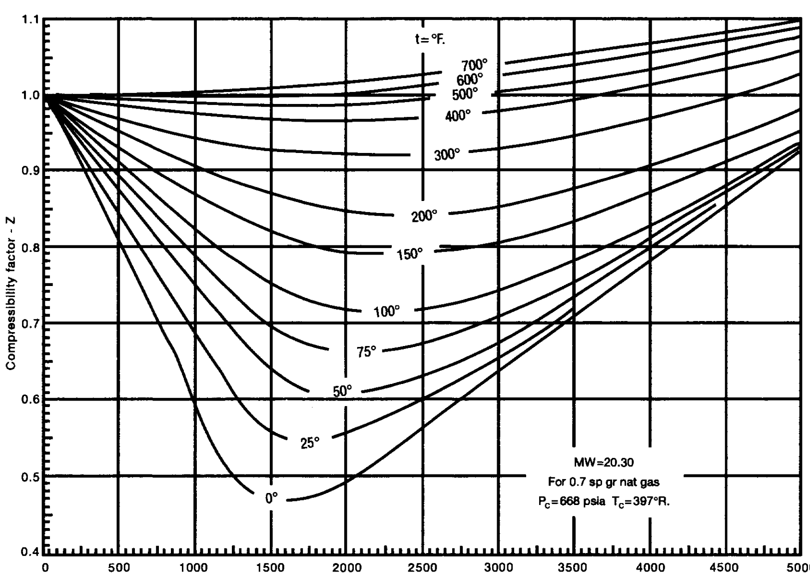

Figure 3-4. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

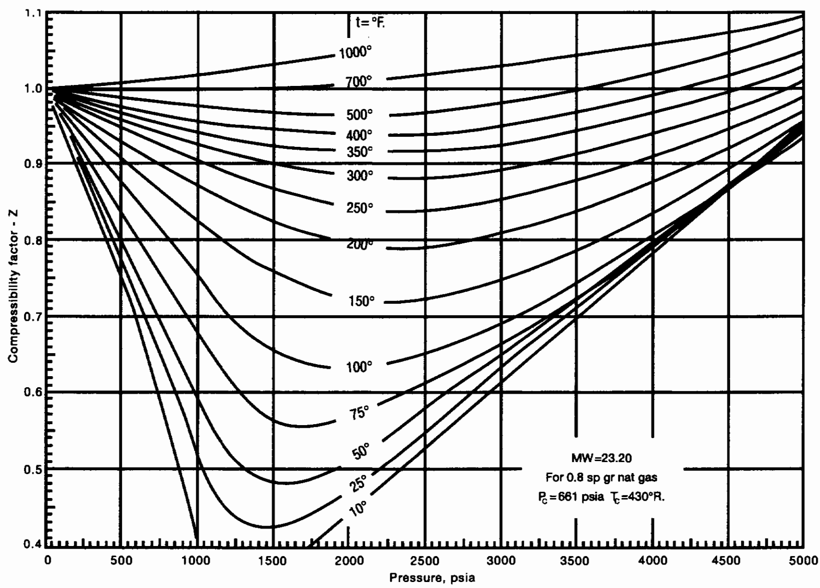

Figure 3-5. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

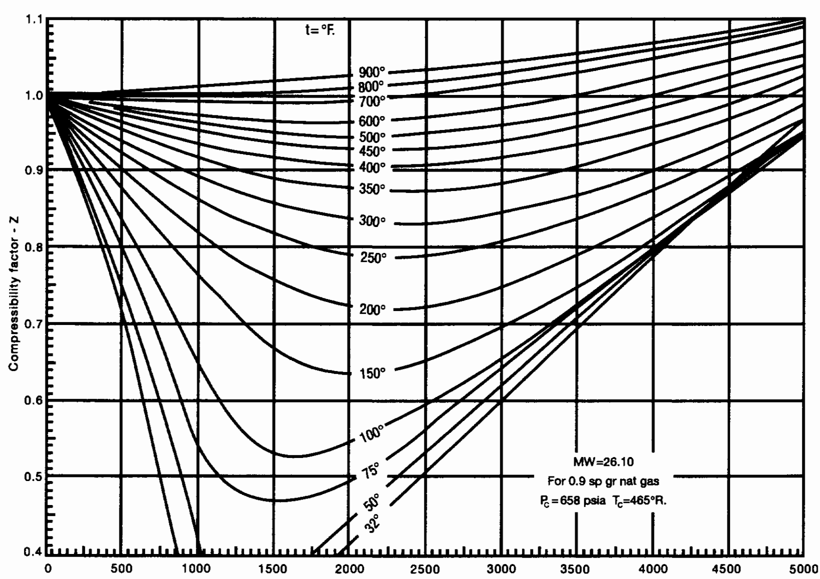

Figure 3-6. Compressibility of low-molecular-weight natural gases (*courtesy of GPSA Data Book*).

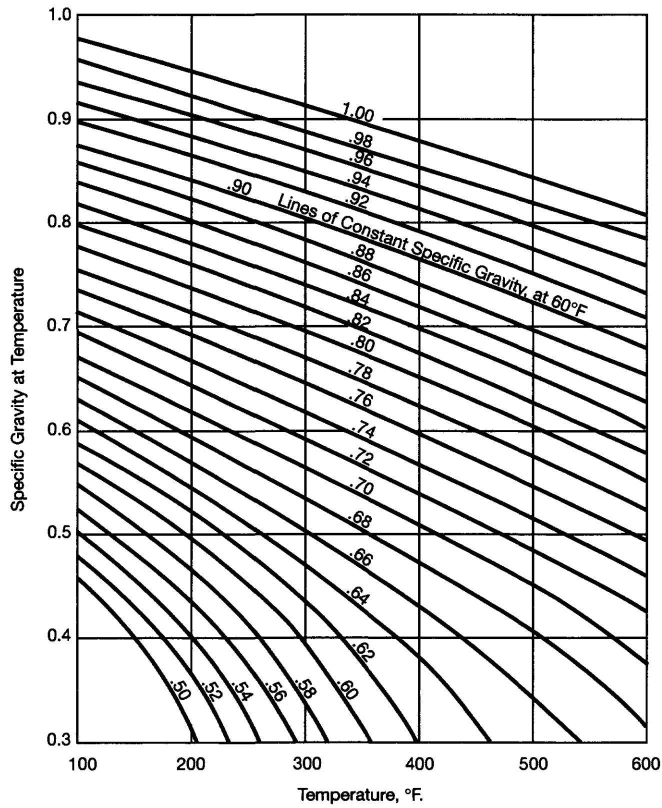

Figure 3-7. Approximate specific gravity of petroleum fractions (*courtesy of GPSA Data Book*).

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hydrocarbon is heated at constant pressure, its specific gravity will *increase* as the lighter hydrocarbons are liberated. The change in the molecular makeup of the fluid is calculated by a "flash calculation," which is described in more detail later.

Viscosity

This property of a fluid indicates its resistance to flow. It is a dynamic property, in that it can be measured only when the fluid is in motion. Viscosity, therefore, is simply the ratio at any shear rate of the shear stress to the shear rate. There are two expressions of viscosity, absolute (or

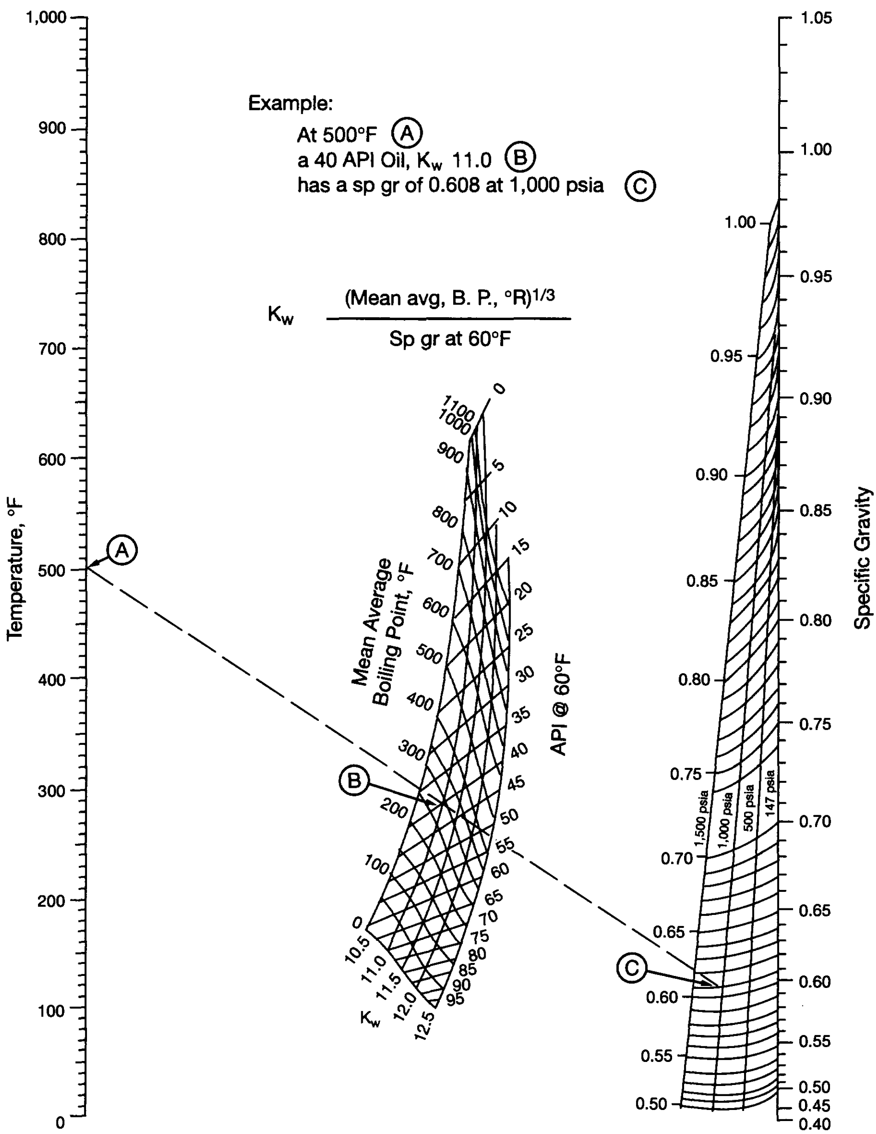

Figure 3-8. Specific gravity of petroleum fractions (*courtesy of Petroleum Refiner: Ritter, Lenoir, and Schweppe 1958*).

dynamic) viscosity, μ, and kinematic viscosity, γ. These expressions are related by the following equation:

$$\gamma = \frac{\mu}{\rho} \tag{3-5}$$

where μ = absolute viscosity, centipoise
γ = kinematic viscosity, centistoke
ρ = density, gram/cm^3

Fluid viscosity changes with temperature. Liquid viscosity decreases with increasing temperature, whereas gas viscosity decreases initially with increasing temperature and then increases with further increasing temperature.

The best way to determine the viscosity of a crude oil at any temperature is by measurement. If the viscosity is known at only one temperature, Figure 3-9 can be used to determine the viscosity at another temperature by striking a line parallel to that for crudes "A," "C," and "D." Care must be taken to assure that the crude does not have its pour point within the temperature range of interest. If it does, its temperature-viscosity relationship may be as shown for Crude "B."

Solid-phase high-molecular-weight hydrocarbons, otherwise known as paraffins, can dramatically affect the viscosity of the crude sample. The cloud point is the temperature at which paraffins first become visible in a crude sample. The effect of the cloud point on the temperature-viscosity curve is shown for Crude "B" in Figure 3-9. This change in the temperature-viscosity relationship can lead to significant errors in estimation. Therefore, care should be taken when one estimates viscosities near the cloud point.

The pour point is the temperature at which the crude oil becomes a solid and ceases to flow, as measured by a specific ASTM procedure (D97). Estimations of viscosity near the pour point are highly unreliable and should be considered accordingly.

In the absence of any laboratory data, correlations exist that relate viscosity and temperature, given the oil gravity. The following equation relating viscosity, gravity, and temperature was developed by Beggs and Robinson after observing 460 oil systems:

$$\mu = 10^x - 1 \tag{3-6}$$

where μ = oil viscosity, cp
T = oil temperature, °F
$x = y(T)^{-1.163}$

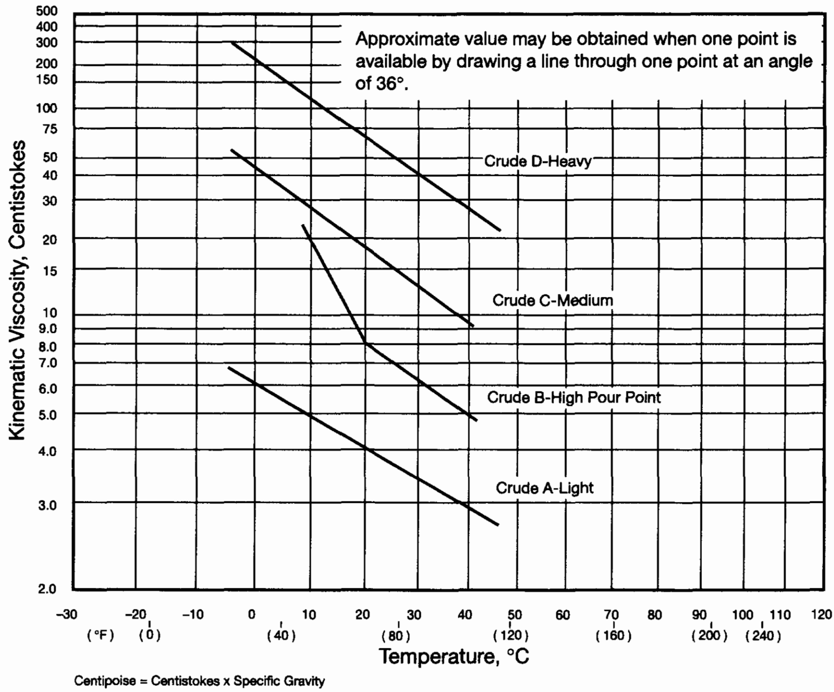

Figure 3-9. Typical viscosity-temperature curves for crude oils (*courtesy of ASTM D-341*).

$$y = 10^z$$
$$z = 3.0324 - 0.02023G$$
$$G = \text{oil gravity, °API}$$

The data set from which this relationship was obtained included a range of between 16° and 58° API and 70°F to 295°F. It has been the author's experience that the correlation tends to overstate the viscosity of the crude oil when dealing in temperature ranges below 100 to 150°F. Figure 3-10 is a graphical representation of another correlation.

The viscosity of produced water depends on the amount of dissolved solids in water as well as the temperature, but for most practical situations it varies from 1.5 to 2 centipoise at 50°F, 0.7 to 1 centipoise at 100°F, and 0.4 to 0.6 centipoise at 150°F.

When an emulsion of oil and water is formed, the viscosity of the mixture may be substantially higher than either the viscosity of the oil or that

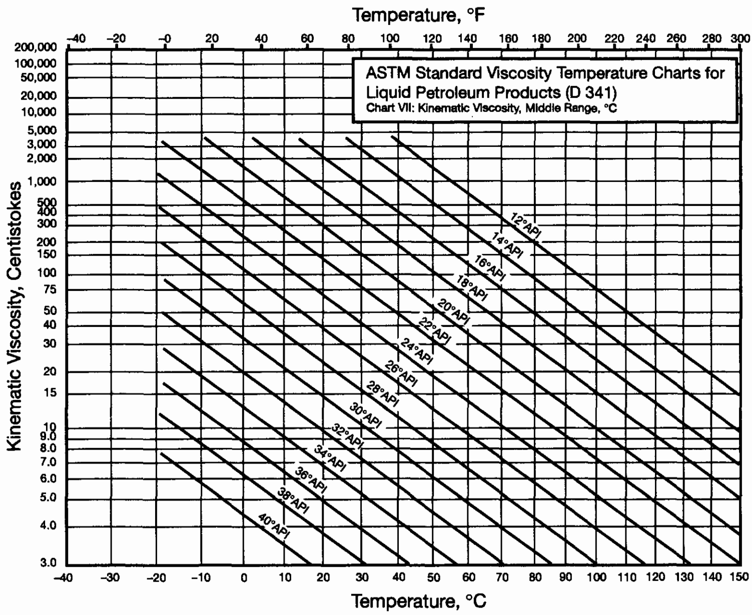

Figure 3-10. Oil viscosity vs. gravity and temperature (*courtesy of Paragon Engineering Services, Inc.*).

of the water taken by themselves. Figure 3-11 shows some experimental data for a mixture of produced oil and water taken from a south Louisiana field. Produced oil and water were mixed vigorously by hand, and viscosity was measured for various percentages of water. For 70% water cut, the emulsion began to break before viscosity readings could be made, and for water cuts greater than this, the oil and water began to separate as soon as the mixing was stopped. Thus, at approximately 70% water cut, it appears as if oil ceases to be the continuous phase and water becomes continuous.

The laboratory data plotted in Figure 3-11 agree closely with a modified Vand's equation assuming a 70% breakover point. This equation is written in the form:

$$\frac{\mu_{eff}}{\mu_c} = 1 + 2.5\,\phi + 10\,\phi^2 \qquad (3-7)$$

where μ_{eff} = effective viscosity
μ_c = viscosity of the continuous phase
ϕ = volume fraction of the discontinuous phase

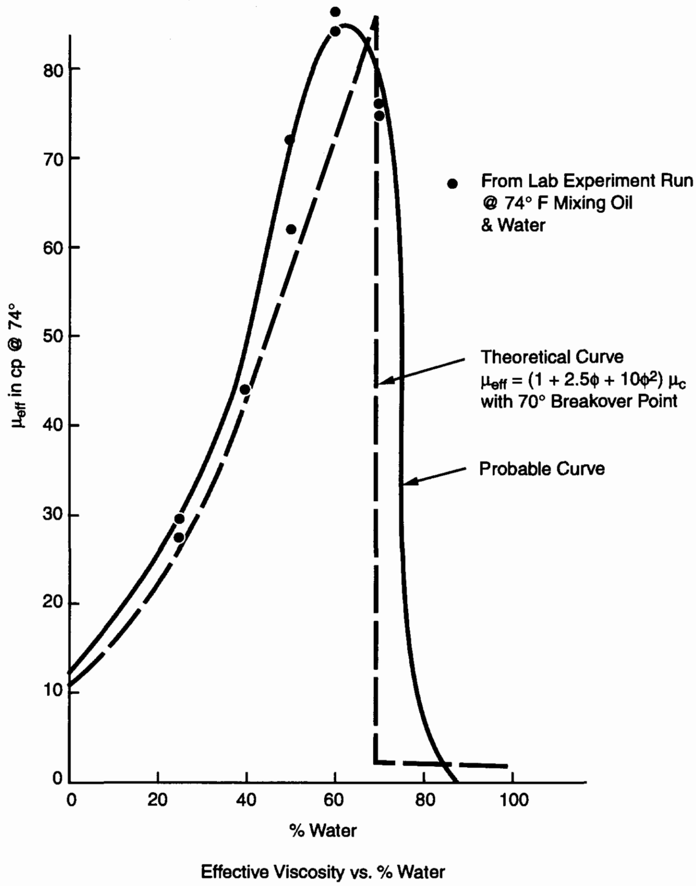

Figure 3-11. Effective viscosity of an oil/water mixture.

Whether the laboratory experiments used to develop this correlation can be accurately compared to the mixing action that occurs in turbulent flow is certainly open to question.

Figure 3-12 can be used to estimate the viscosity of a hydrocarbon gas at various conditions of temperature and pressure if the specific gravity of the gas at standard conditions is known.

FLASH CALCULATIONS

The amount of hydrocarbon fluid that exists in the gaseous phase or the liquid phase at any point in the process is determined by a flash calcula-

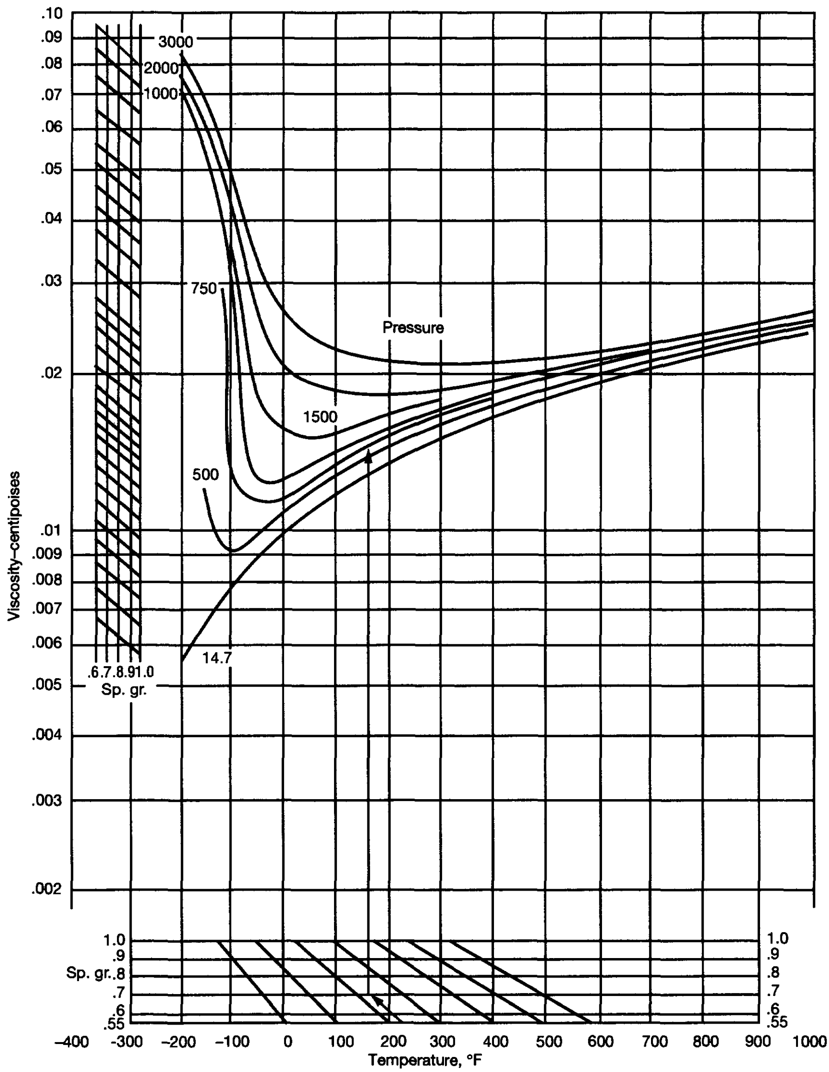

Figure 3-12. Hydrocarbon gas viscosity (*courtesy of GPSA Engineering Data Book*).

tion. As explained in Chapter 2, for a given pressure and temperature, each component of the hydrocarbon mixture will be in equilibrium. The mole fraction of the component in the gas phase will depend not only on pressure and temperature, but also on the partial pressure of that component. Therefore, the amount of gas depends upon the total composition of

the fluid as the mole fraction of any one component in the gas phase is a function of the mole fraction of every other component in this phase.

This is best understood by assigning an equilibrium "K" value to each component. The K value is a strong function of temperature and pressure and of the composition of the vapor and liquid phase. It is defined as:

$$K_N = \frac{V_N / V}{L_N / L} \tag{3-8}$$

where K_N = constant for component N at a given temperature and pressure
V_N = moles of component N in the vapor phase
V = total moles in the vapor phase
L_N = moles of component N in the liquid phase
L = total moles in the liquid phase

The Gas Processors Suppliers Association (GPSA) presents graphs of K values for the important components in a hydrocarbon mixture such as those in Figures 3-13 to 3-25. The K values are for a specific "convergence" pressure to consider the composition of the vapor and liquid phase. There is a procedure in the GPSA Engineering Data Book for calculating convergence pressure based on simulating the fluid as a binary system with the lightest hydrocarbon component, which makes up at least 0.1 mole percent in the liquid and a pseudo heavy component having the same weight average critical temperature as the remaining heavier hydrocarbons. The convergence pressure is then read from a graph of convergence pressure versus operating temperature for common pseudo-binaries.

In most oil field applications the convergence pressure will be between 2,000 and 3,000 psia, except at very low pressures, between 500 and 1,500 psia are possible. If the operating pressure is much less than the convergence pressure, the equilibrium constant is not greatly affected by the choice of convergence pressure. Therefore, using a convergence pressure of 2,000 psia is a good first approximation for most flash calculations. Where greater precision is required, the convergence pressure should be calculated.

If K_N for each component and the ratio of total moles of vapor to total moles of liquid (V/L) are known, then the moles of component N in the vapor phase (V_N) and the moles in the liquid phase (L_N) can be calculated from:

(text continued on page 85)

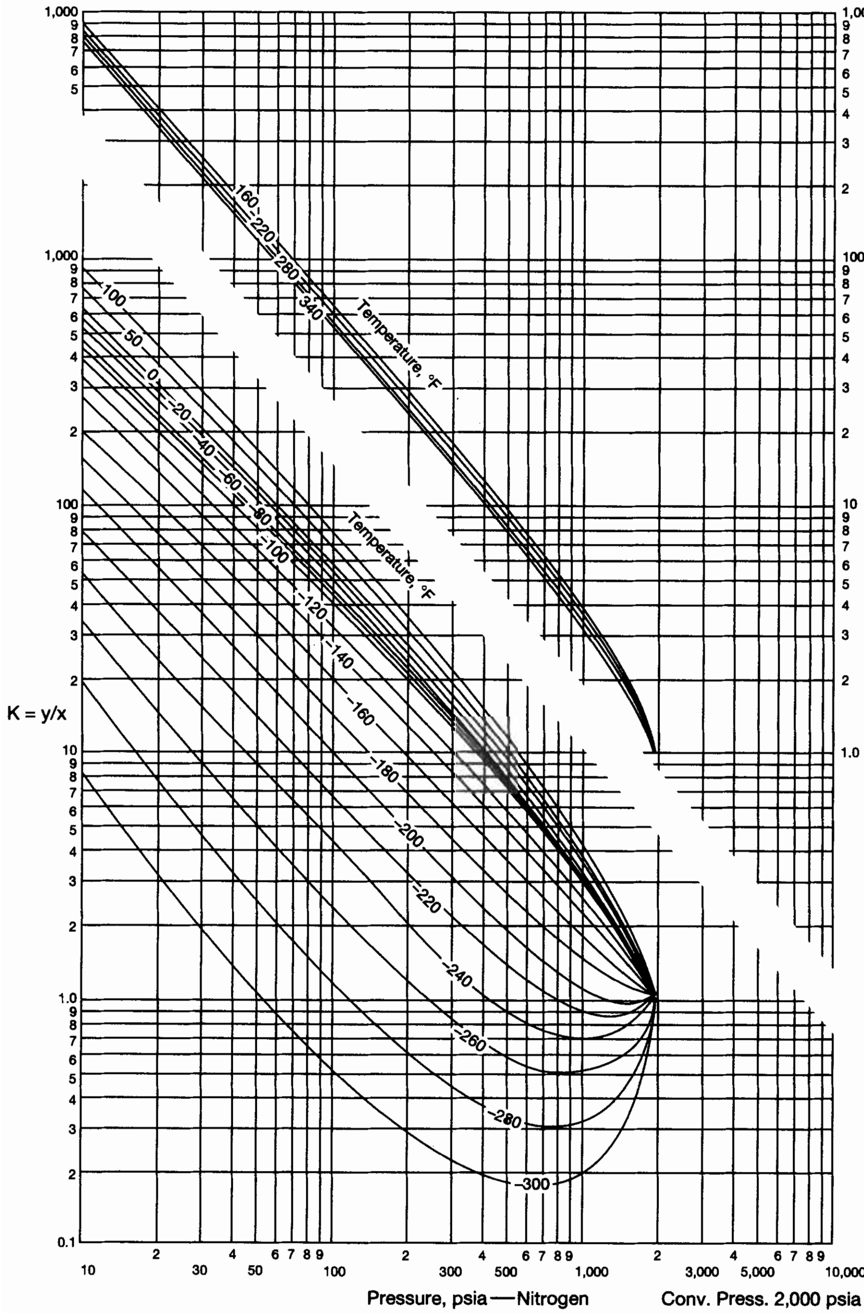

Figure 3-13. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

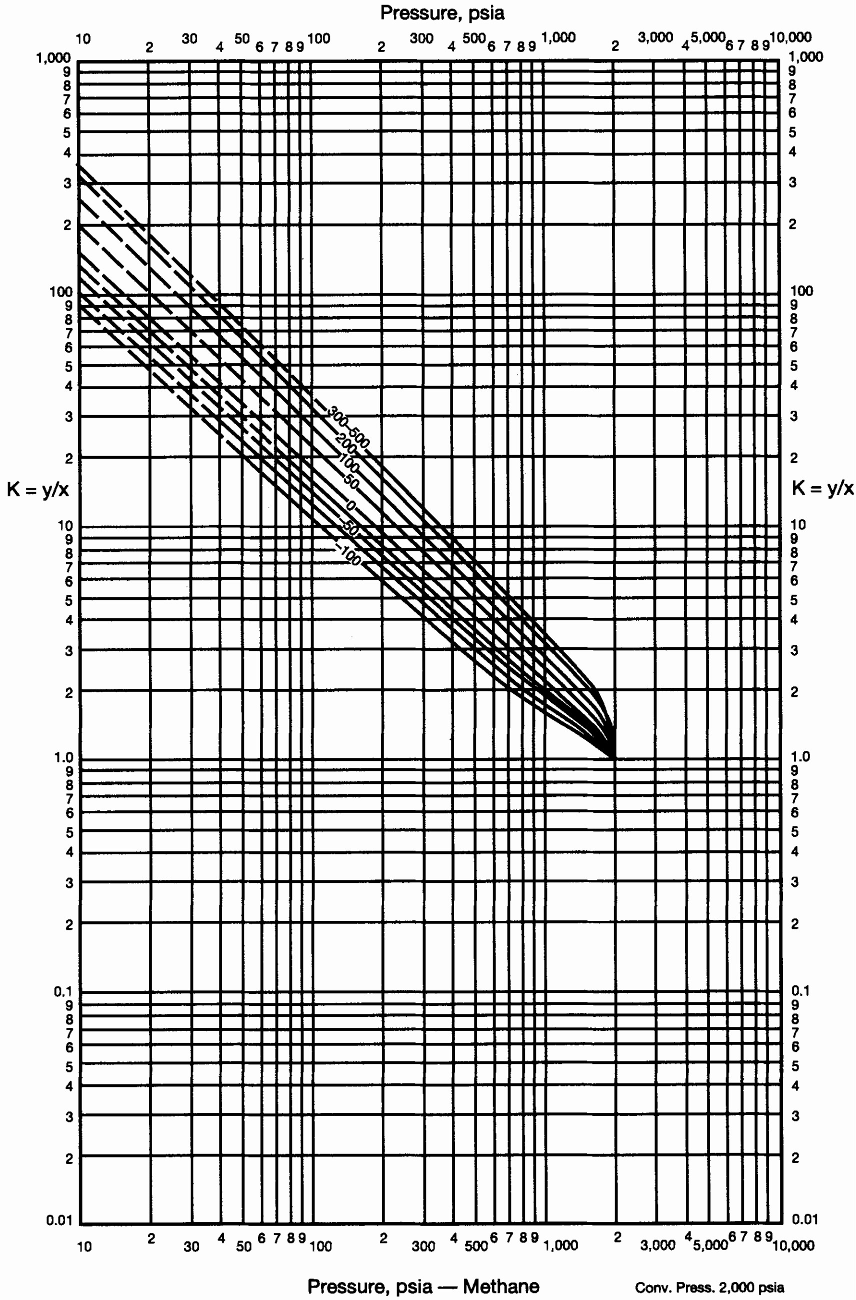

Figure 3-14. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

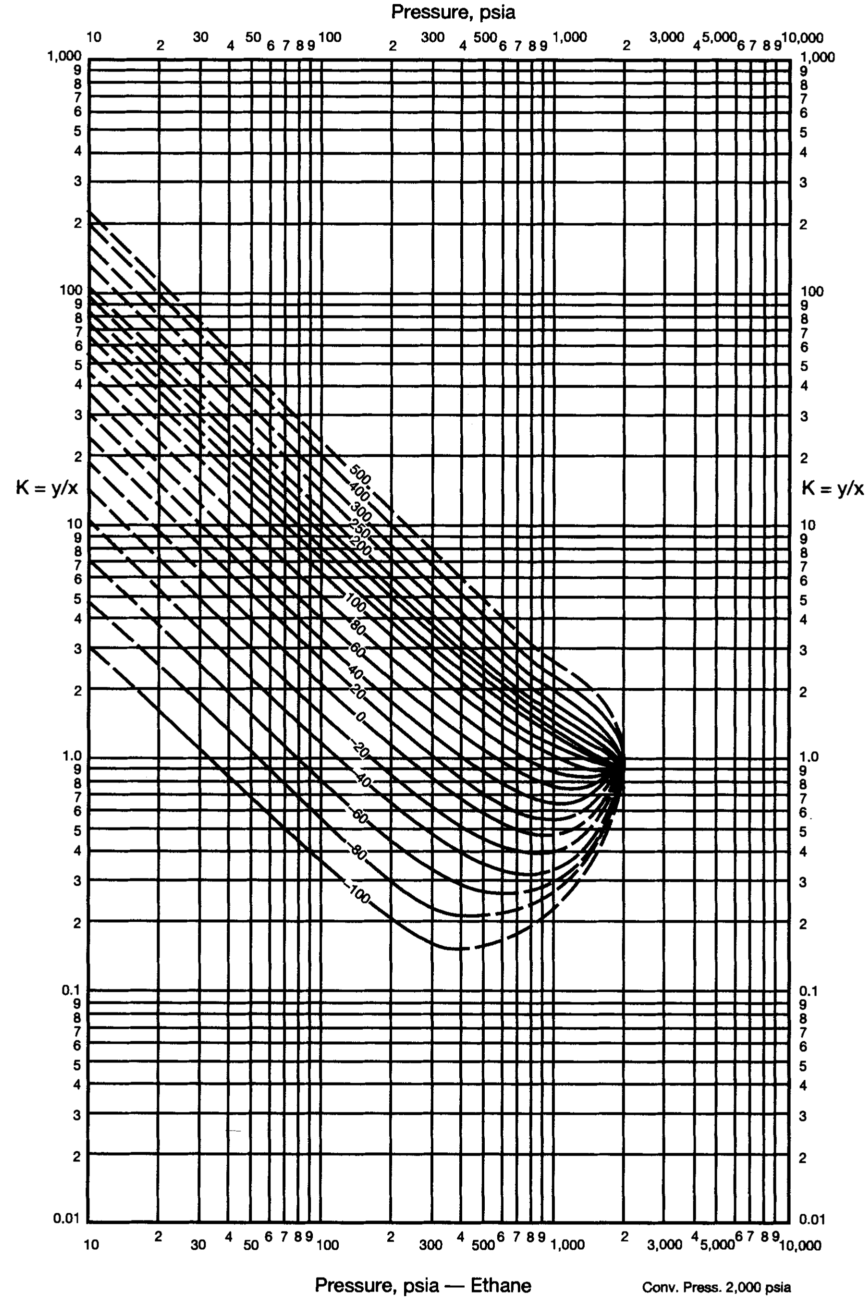

Figure 3-15. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

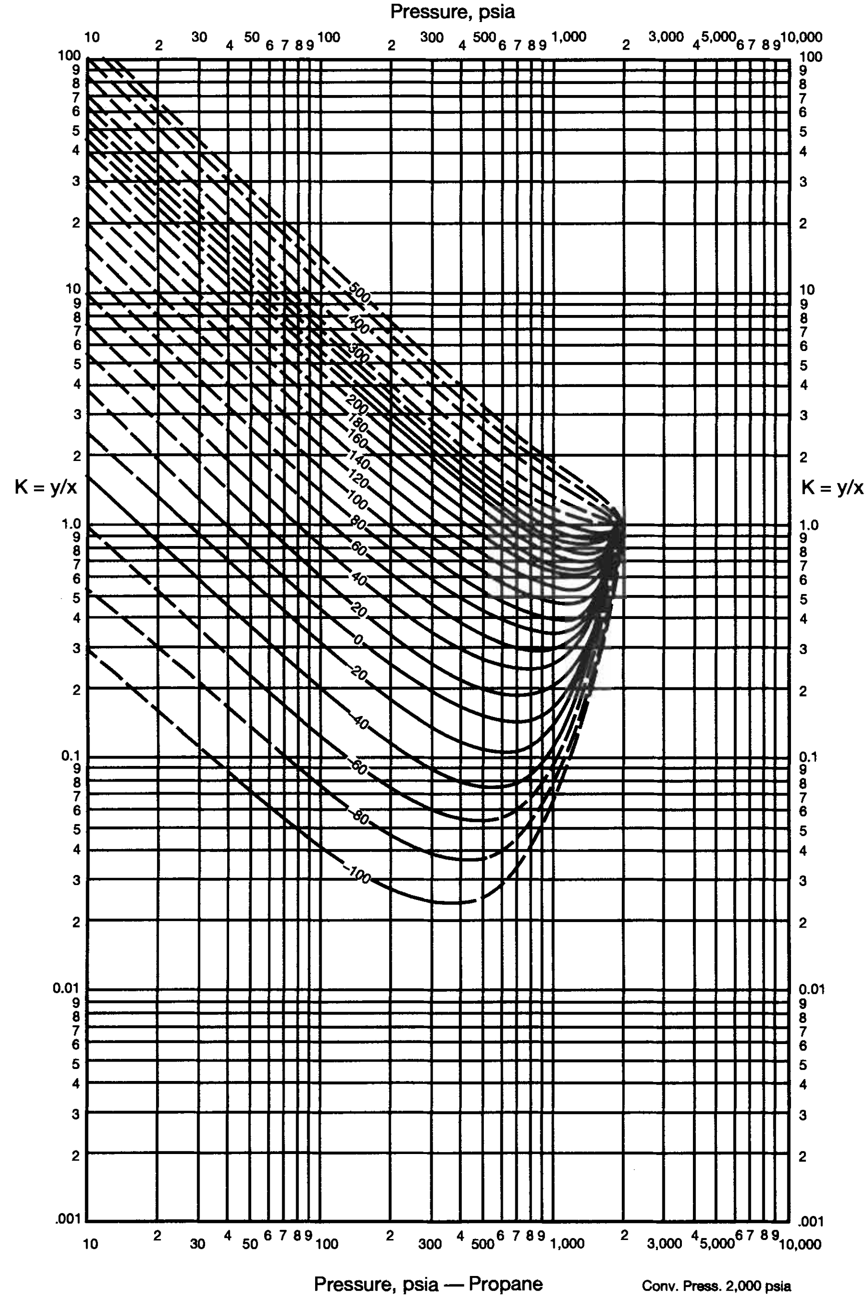

Figure 3-16. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

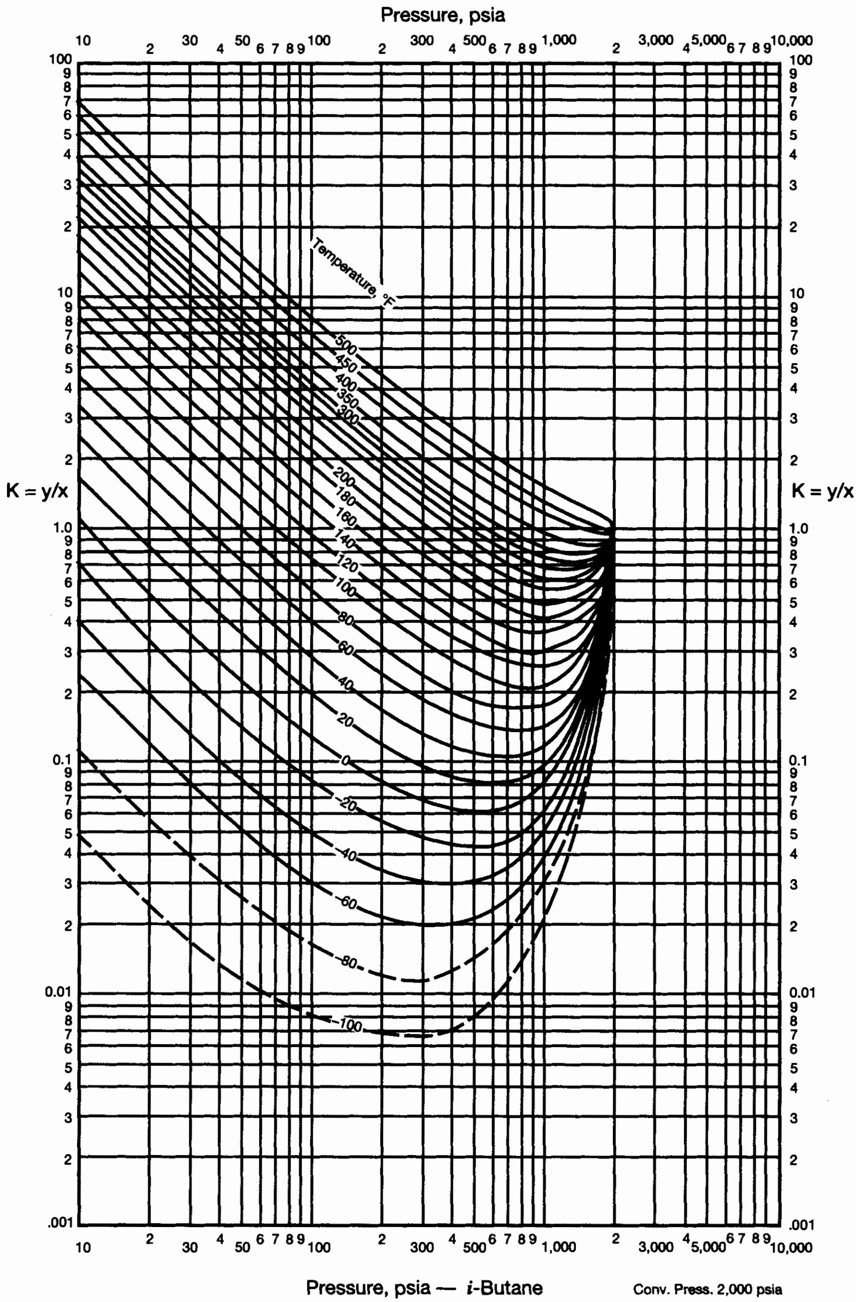

Figure 3-17. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

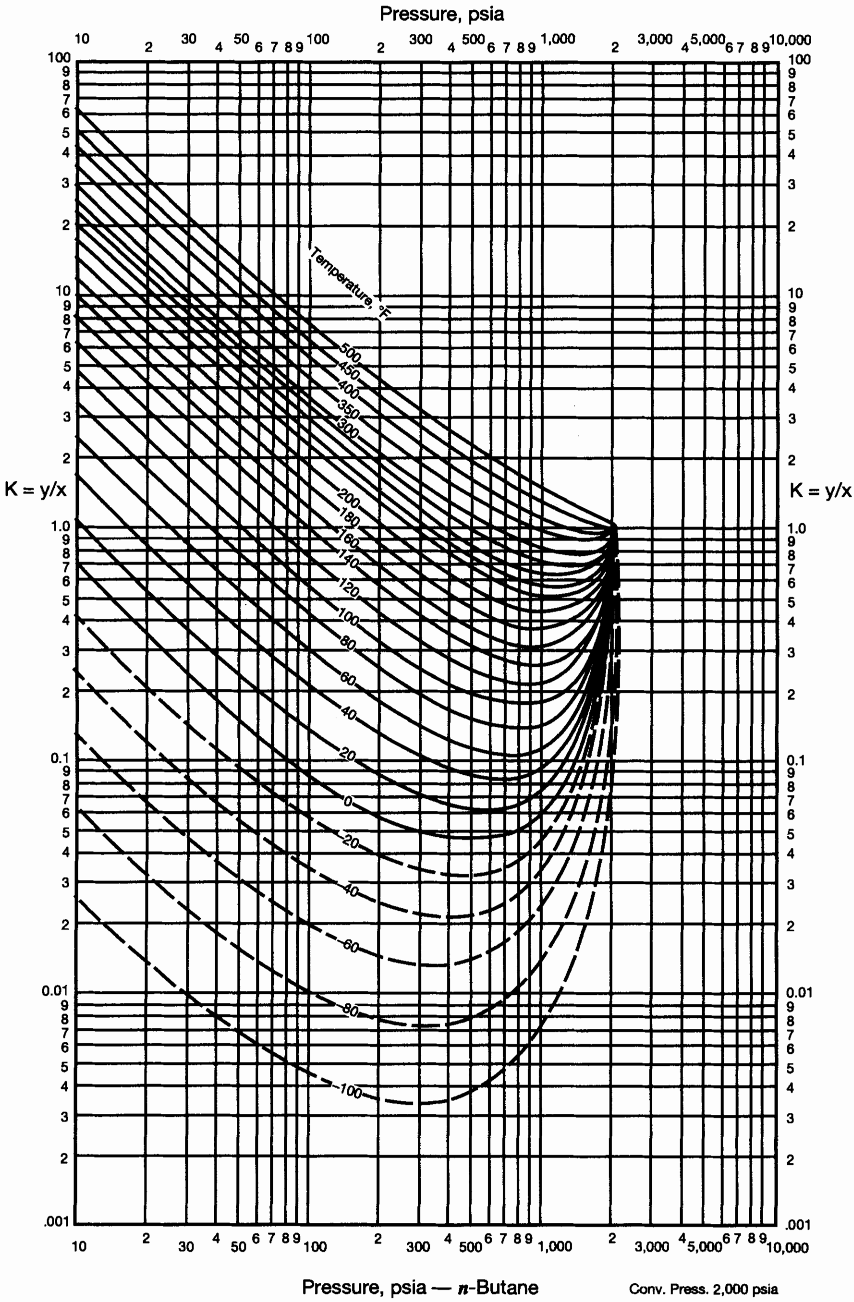

Figure 3-18. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

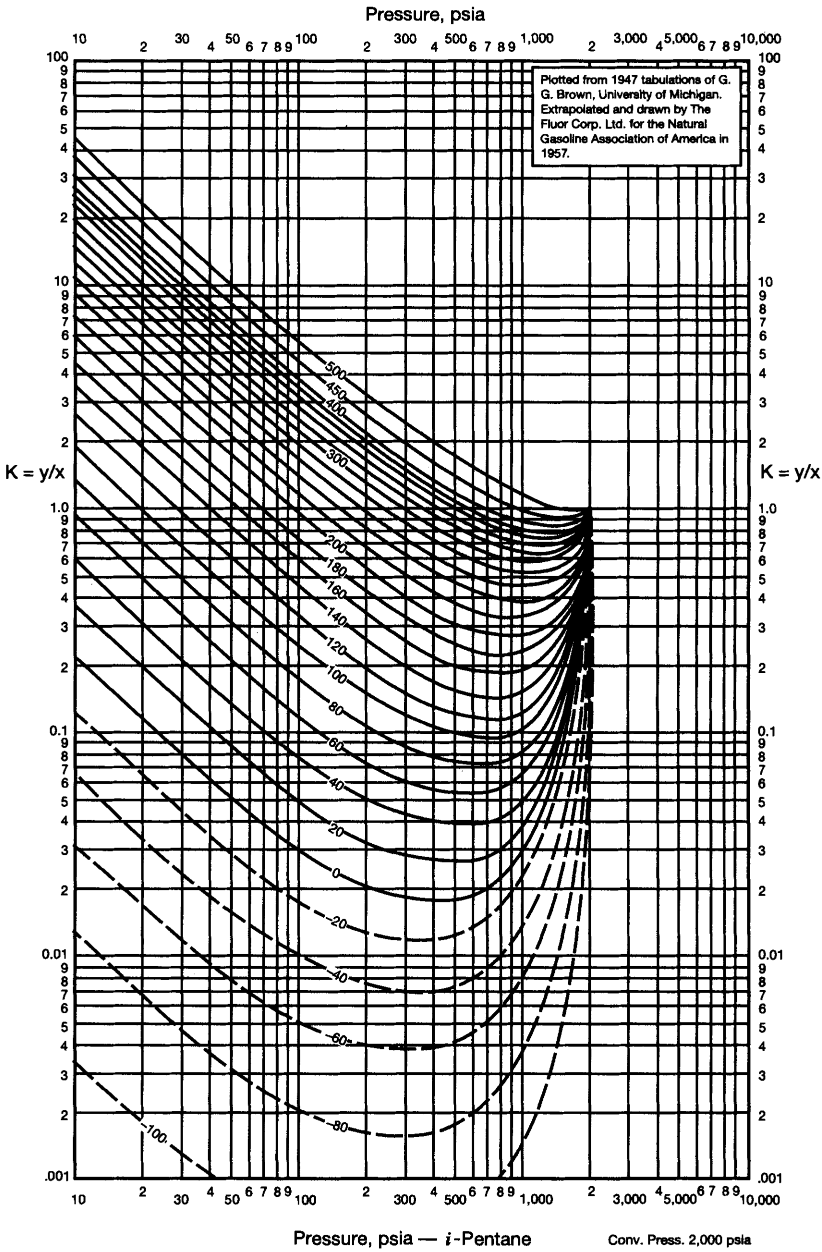

Figure 3-19. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

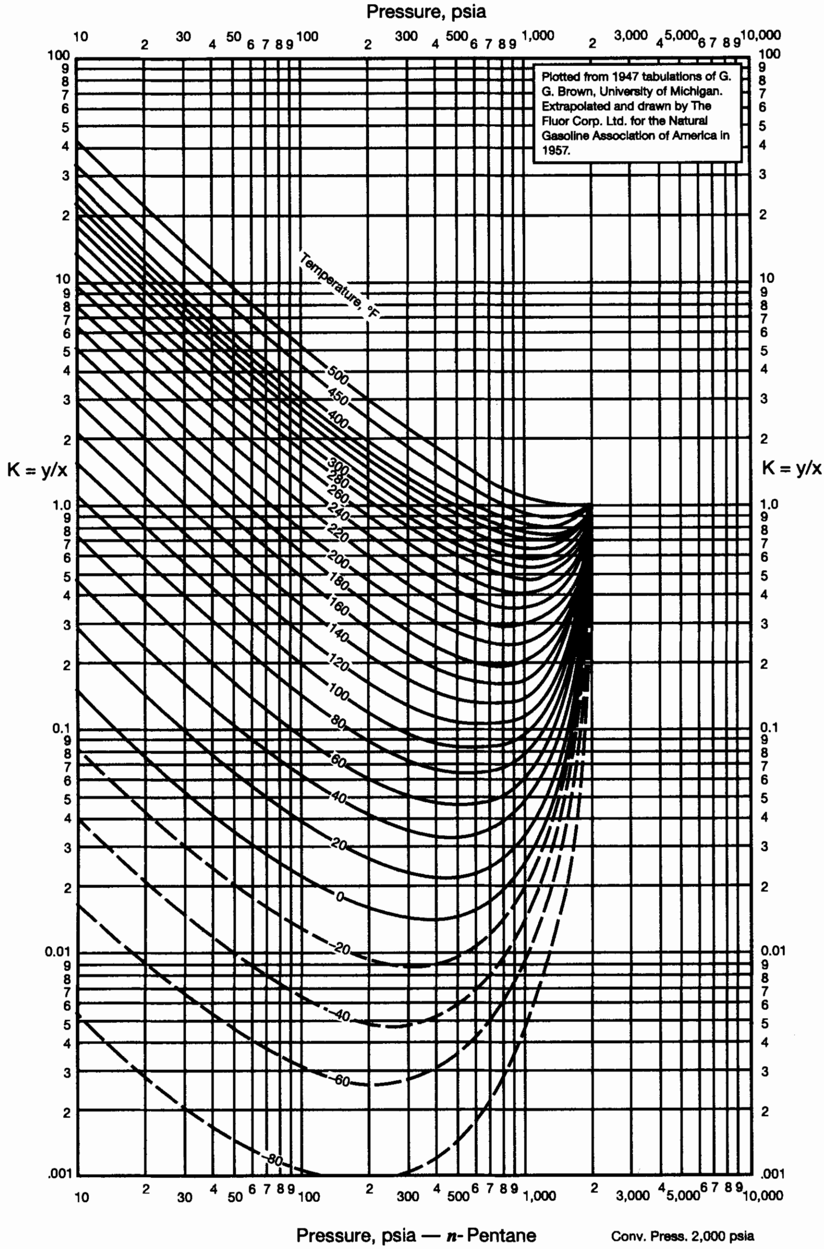

Figure 3-20. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

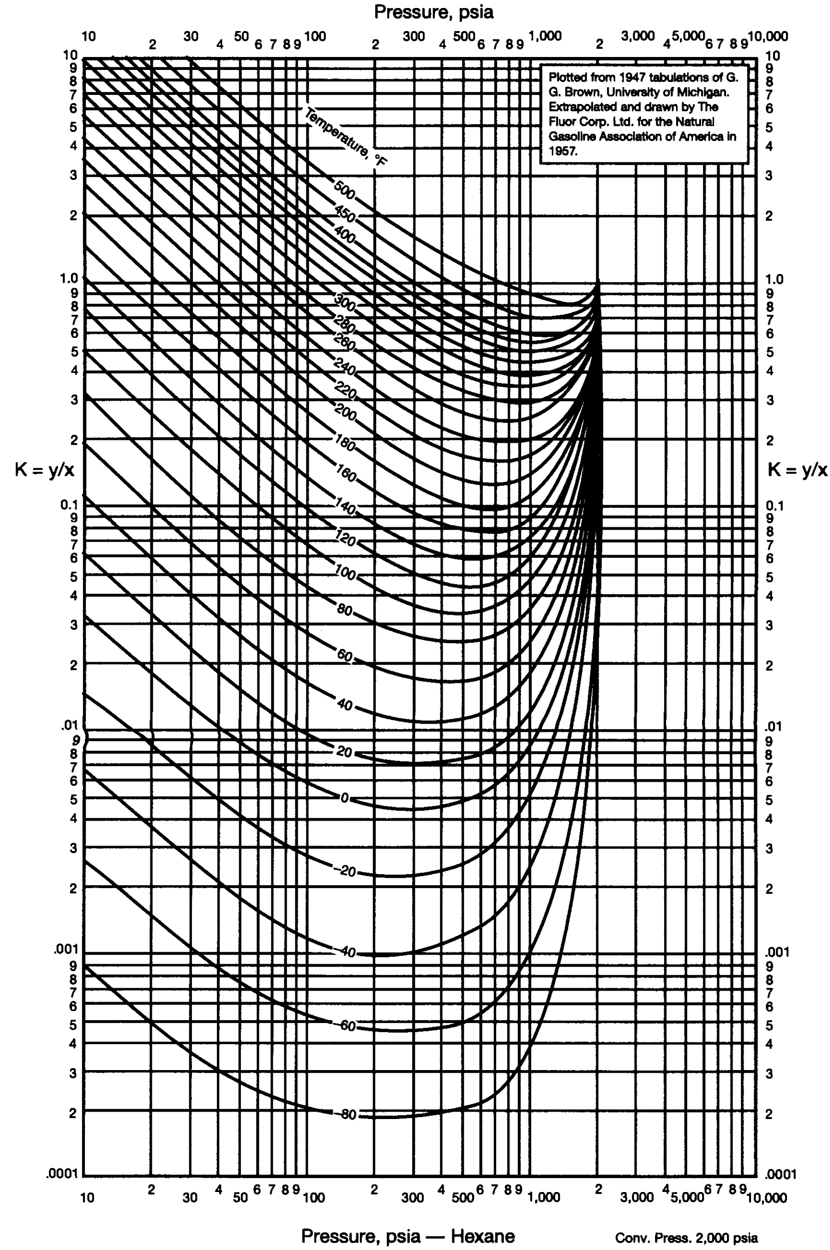

Figure 3-21. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

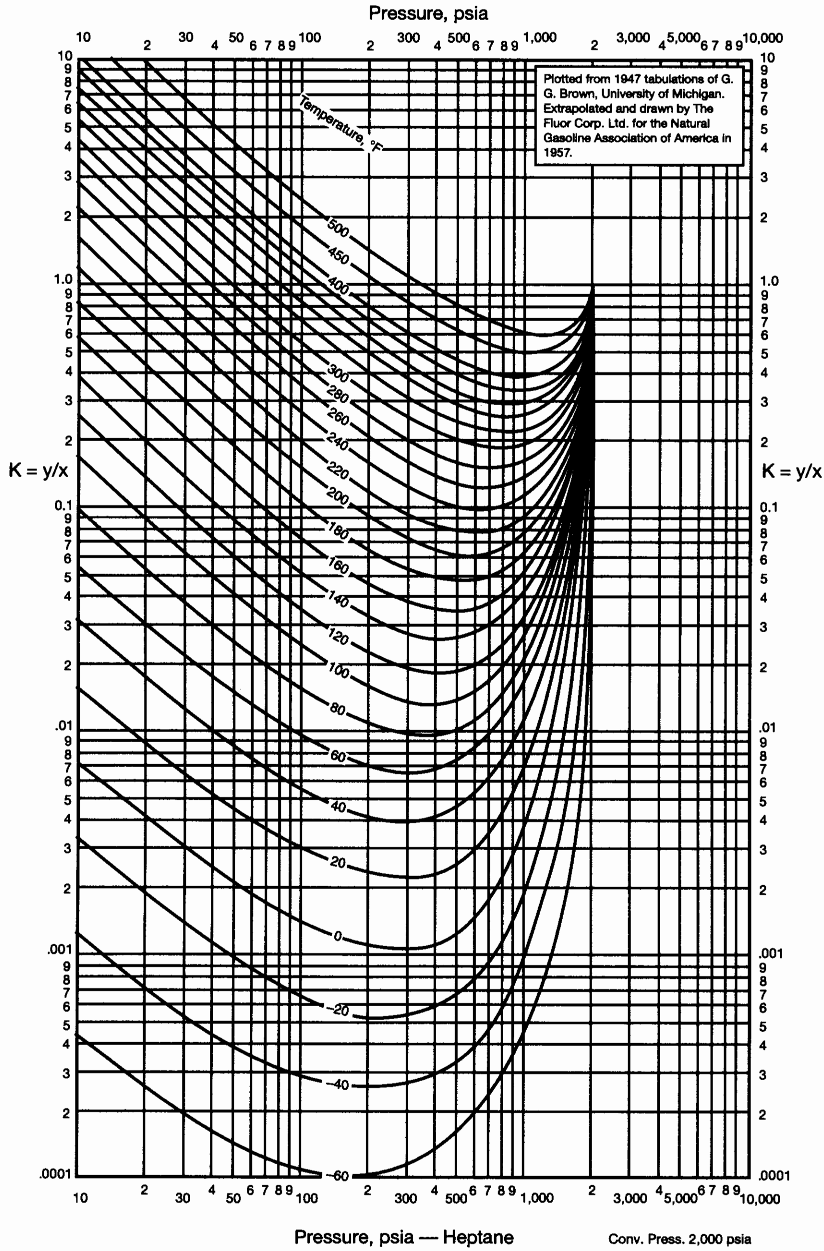

Figure 3-22. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

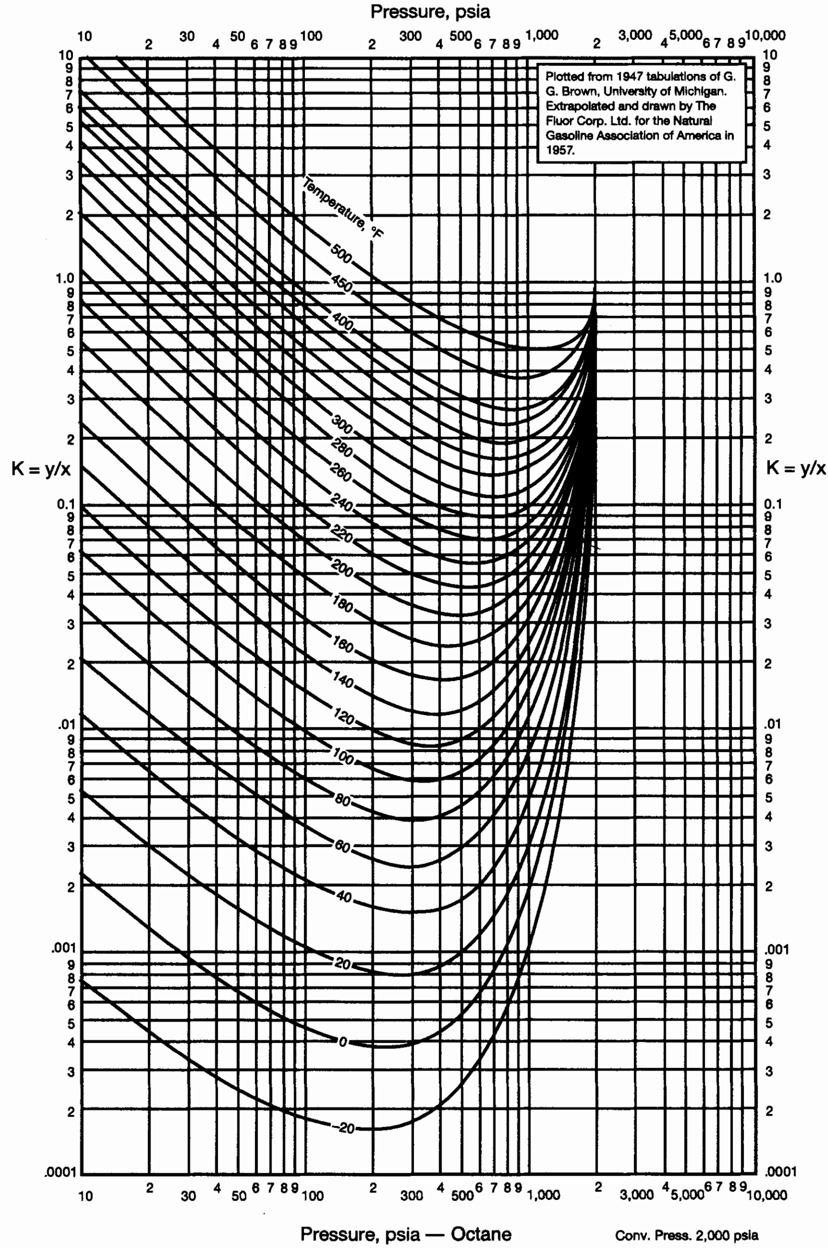

Figure 3-23. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

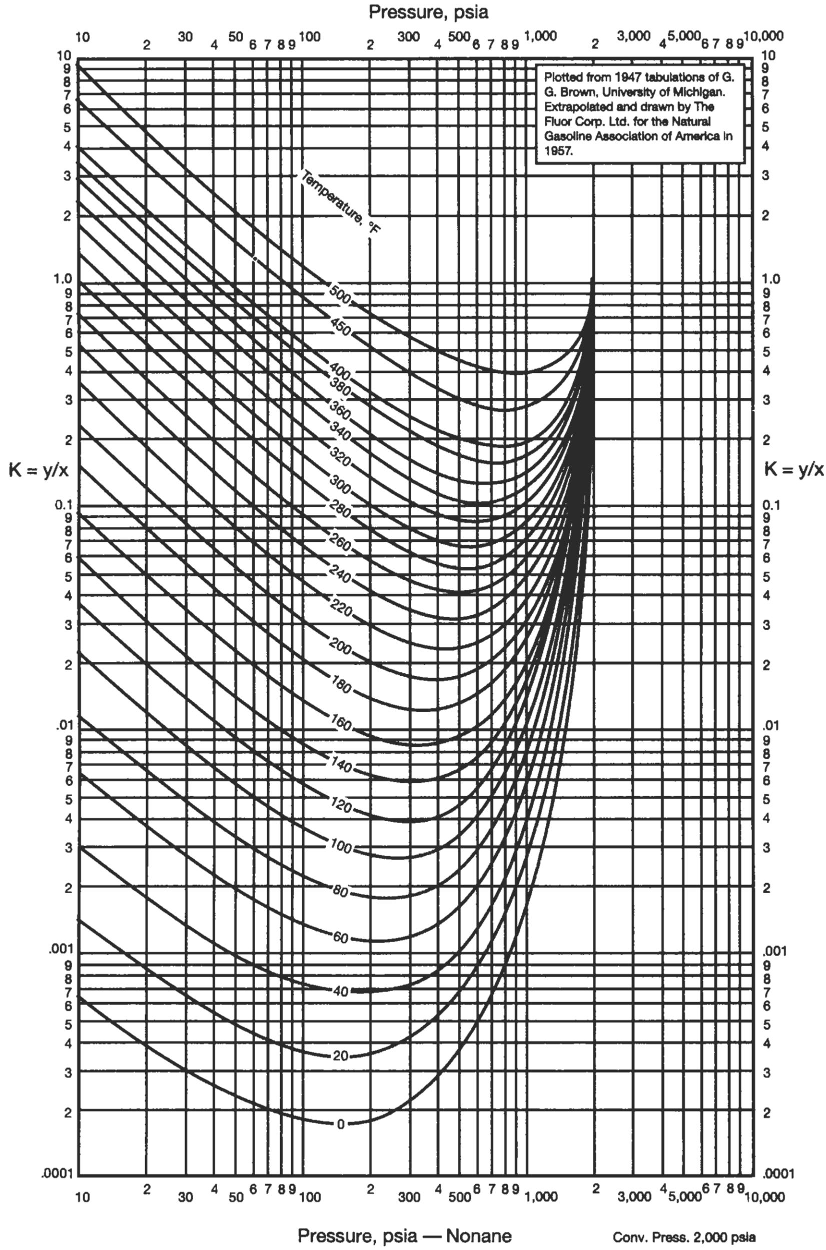

Figure 3-24. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

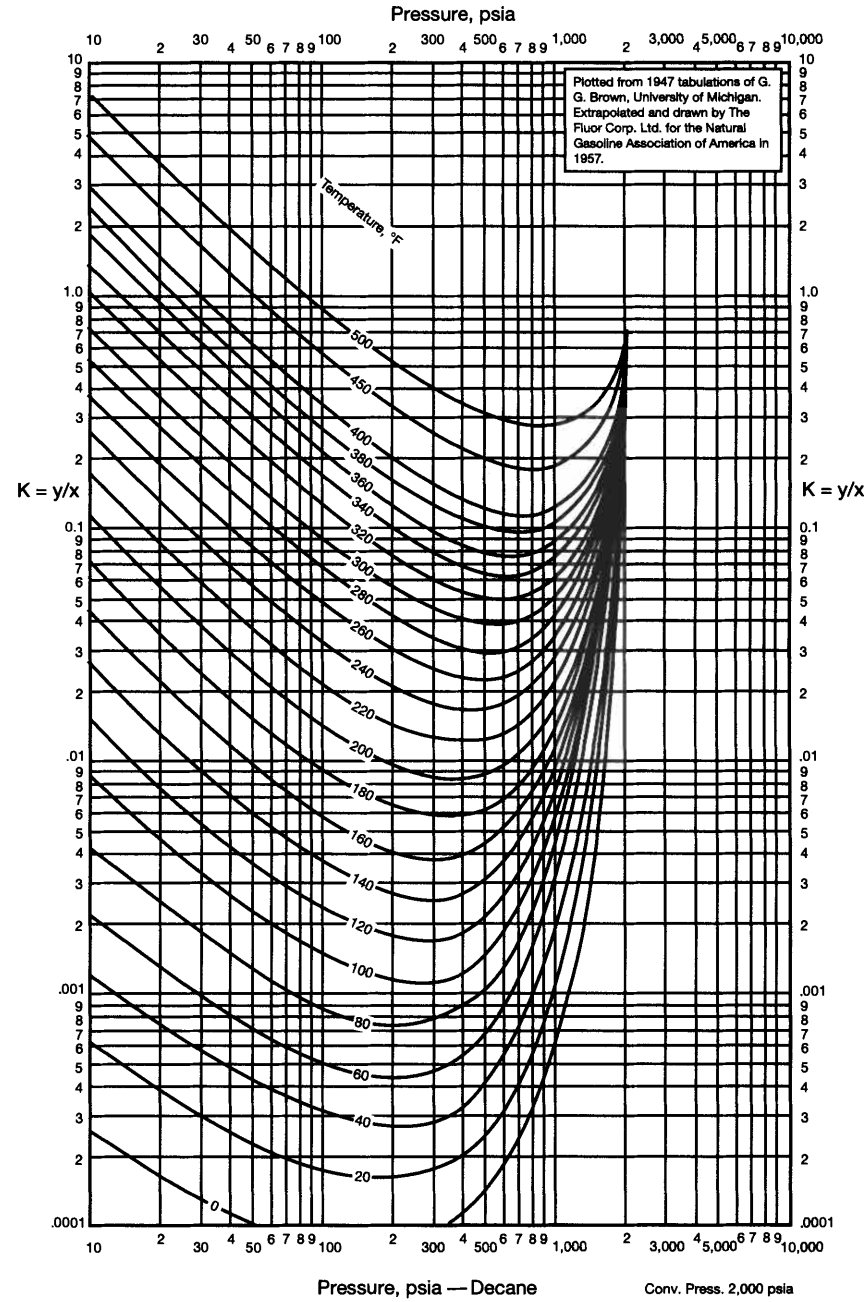

Figure 3-25. "K" values for hydrocarbon mixtures (*courtesy of GPSA Engineering Data Book*).

(text continued from page 71)

$$V_N = \frac{K_N F_N}{\dfrac{1}{(V/L)} + K_N} \tag{3-9}$$

$$L_N = \frac{F_N}{K_N (V/L) + 1} \tag{3-10}$$

where F_N = total moles of component N in the fluid

Derivations of Equations 3-9 and 3-10

$$K_N = \left(\frac{V_N}{L_N}\right)\frac{1}{(V/L)}$$

$$L_N = \left(\frac{V_N}{K_N}\right)\frac{1}{(V/L)}$$

$$F_N = L_N + V_N$$

$$F_N = \left(\frac{V_N}{K_N}\right)\frac{1}{(V/L)} + V_N$$

$$F_N = \left[\frac{1}{K_N (V/L)} + 1\right] V_N$$

$$V_N = \frac{K_N F_N}{\dfrac{1}{(V/L)} + K_N}$$

$$V_N = L_N K_N (V/L)$$

$$F_N = L_N + L_N K_N (V/L)$$

$$L_N = \frac{F_N}{K_N (V/L) + 1}$$

To solve either Equation 3-9 or 3-10, it is necessary to first know the quantity (V/L), but since both V and L are determined by summing V_N

and L_N, it is necessary to use an iterative solution. This is done by first estimating (V/L), calculating V_N and L_N for each component, summing up to obtain the total moles of gas (V) and liquid (L), and then comparing the calculated (V/L) to the assumed value. In doing this procedure, it is helpful to use the relationship:

$$L = \frac{F}{1 + (V/L)} \tag{3-11}$$

Derivation of Equation 3-11

$$F = V + L$$

$$V = (V/L)L$$

$$F = (V/L)L + L = L[(V/L) + 1]$$

$$L = \frac{F}{1 + (V/L)}$$

Once an assumed value of V/L is made, it is easy to calculate the corresponding assumed value of L. This is best illustrated by the example in Table 3-1. The mole fraction (Column 2) for each component is given from test data. Column 3 is determined from the graphs for K_N, assuming

Table 3-1
Flash Calculation at 1,000 psia and 100°F

(1) Component	(2) Mole Fraction Percent	(3) K_N	(4) V/L = 1.5, L = 40, L_N	(5) V/L = 0.5, L = 66.7, L_N	(6) V/L = 1, L = 50, L_N	(7) V_N
CO_2	0.22	1.88*	0.06	0.11	0.08	0.14
N_2	0.09	4.00	0.01	0.03	0.02	0.07
Methane	63.35	2.80	12.18	26.40	16.67	46.68
Ethane	4.21	0.96	1.73	2.84	2.15	2.06
Propane	2.09	0.38	1.33	1.76	1.51	0.58
I-Butane	0.68	0.22	0.51	0.61	0.56	0.12
N-Butane	1.08	0.18	0.85	0.99	0.92	0.16
I-Pentane	0.47	0.10	0.41	0.45	0.43	0.04
N-Pentane	0.38	0.09	0.33	0.36	0.35	0.03
Hexane	1.36	0.05	1.20	1.33	1.30	0.06
Heptane+	26.07	0.006**	25.84	25.99	25.91	0.16
	100.00		44.45	60.87	49.90	50.10
					100.00	

* *Calculated as* $K_{CO_2} = (K_{C_1} + K_{C_2})^{1/2}$
** *Simulated as decane*

a convergence pressure of 2,000 psia. Column 4 is derived from Equation 3-10, assuming F = 100 moles and V/L = 1.5. That is, L = 40 moles. With this assumption it is calculated that L_N = 44.45 moles and this is plotted in Figure 3-26 as point "1."

Another assumption is then made that V/L = 0.5 (i.e., L = 66.7 moles) and in Column 5, L is calculated to be 60.87. This is plotted as point "2." From Figure 3-26, point "3," which represents the intersection of assumed and calculated values, indicates an L ≅ 50, which corresponds to a V/L ≅ 1.0. This is tabulated in Column 6. It can be seen that L_N is calculated as 49.9. Column 7, which characterizes the composition of the gas stream, is obtained by the difference between Column 6 and Column 2.

CHARACTERIZING THE FLOW STREAM

Once a flash calculation is made and the molecular composition of the liquid and gas components have been determined, it is possible to determine the properties and flow rates of both the gas and the liquid streams.

Molecular Weight of Gas

The molecular weight of the stream is calculated from the weighted average gas molecular weight given by:

$$MW = \frac{\Sigma [V_N \times (MW)_N]}{V} \qquad (3-12)$$

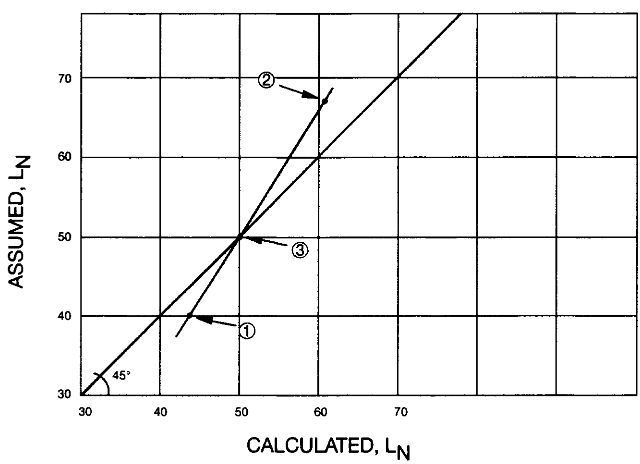

Figure 3-26. Interpolation of flash calculations results.

The molecular weight of the gas stream of Table 3-1 is calculated in Table 3-2. Column 2 lists the molecular weight of the components from standard reference sources. Column 3 lists the number of moles of each component for 100 moles of feed. This is the same as Column 7 in Table 3-1. Column 4 is derived from Column 2 times Column 3. The molecular weight of the gas is:

$$MW = \frac{911.5}{50.1} = 18.19$$

The gas specific gravity can be determined from the molecular weight from Equation 3-2 as shown in Table 3-2.

$$S = \frac{18.19}{29} = 0.63$$

Table 3-2
Gas Flow Characterization

(1) Component	(2) MW_N	(3) V_N Moles	(4) $V_N \times MW_N$
CO_2	44.01	0.14	6.2
N_2	28.01	0.07	2.0
Methane	16.04	46.68	748.7
Ethane	30.07	2.06	61.9
Propane	44.10	0.58	25.6
I-Butane	58.12	0.12	7.0
N-Butane	58.12	0.16	9.3
I-Pentane	72.15	0.04	2.9
N-Pentane	72.15	0.03	2.2
Hexane	86.18	0.06	5.2
Heptane+	253.00*	0.16	40.5
		50.10	911.5

* *From PVT analysis of feed stream.*

Gas Flow Rate

If the flow rate of the inlet stream is known in moles per day then the number of moles per day of gas flow can be determined from:

$$V = \frac{F}{1 + \frac{1}{(V/L)}} \tag{3-13}$$

where V = gas flow rate, moles/day
F = total stream flow rate, moles/day
L = liquid flow rate, moles/day

Derivation of Equation 3-13

$$F = V + L$$

$$V = (V/L)L$$

$$F = V + \frac{V}{(V/L)} = V\left[1 + \frac{1}{(V/L)}\right]$$

$$V = \frac{F}{1 + \frac{1}{(V/L)}}$$

Once the mole flow rate of gas is known, then the flow rate in standard cubic feet can be determined by recalling that one mole of gas occupies 380 cubic feet at standard conditions. Therefore:

$$Q_g = \frac{380V}{1,000,000} \tag{3-14}$$

where Q_g = gas flow rate, MMscfd

Assuming a feed flow rate of 10,000 moles per day for the stream being flashed in Table 3-1:

$$V = \frac{10,000}{1 + \frac{1}{50.1/49.9}} = 5,010 \text{ moles/day}$$

$$Q_g = \frac{380}{1,000,000}(5,010) = 1.90 \text{ MMscfd}$$

Liquid Molecular Weight

The molecular weight of the liquid stream is calculated from the weighted average liquid component molecular weight given by:

$$MW = \frac{\Sigma[L_N \times (MW)_N]}{L} \tag{3-15}$$

This is calculated in Table 3-3. Column 2 is as in Table 3-2 and Column 3 is the liquid stream composition for 100 moles of feed as calculat-

ed in Table 3-1, Column 7. Column 4 is Column 2 times Column 3 and represents the weight of each component in the liquid stream. The molecular weight of the liquid is:

$$MW = \frac{7,212}{49.90} = 145$$

Table 3-3
Liquid Flow Characterization

(1) Component	(2) MW_N	(3) L_N	(4) $L_N \times (MW_N)$	(5) $(S.G._N)$	(6) $\frac{L_N \times (MW)_N}{(S.G._N)}$
CO_2	44.01	0.08	3	0.83**	4
N_2	28.01	0.02	1	0.81**	1
Methane	16.04	16.67	267	0.30**	891
Ethane	30.07	2.15	65	0.36**	179
Propane	44.10	1.51	67	0.51**	131
I-Butane	58.12	0.56	33	0.56**	58
N-Butane	58.12	0.92	53	0.58**	92
I-Pentane	72.15	0.43	31	0.62	50
N-Pentane	72.15	0.35	25	0.63	40
Hexane	86.18	1.30	112	0.66	170
Heptane+	253.00*	25.91	6,555	0.86*	7,622
		49.90	7,212		9,238

* *From PVT analysis of feed stream.*
** *Pseudo value at saturation pressure.*

Specific Gravity of Liquid

Remembering that the weight of each component is the number of moles of that component times its molecular weight (pounds = MW × moles), the specific gravity of the liquid is given by:

$$S.G. = \frac{\Sigma\,[L_N \times (MW)_N]}{\Sigma\left[\frac{L_N \times (MW)_N}{(S.G.)_N}\right]} \qquad (3-16)$$

Derivation of Equation 3-16

ρ in lb/ft^3, volume in ft^3

$$S.G. = \frac{\rho}{62.4}$$

$$\rho = \frac{\Sigma\, Pounds_N}{\Sigma\, Volume_N}$$

$$S.G. = \frac{1}{62.4} \frac{\Sigma \text{ Pounds}_N}{\Sigma \text{ Volume}_N}$$

$$\text{Volume}_N = \frac{\text{Pounds}_N}{\rho_N}$$

$$\text{Volume}_N = \frac{\text{Pounds}_N}{62.4\,(S.G.)_N}$$

$$\text{Volume}_N = \frac{1}{62.4} \sum \frac{\text{Pounds}_N}{(S.G.)_N}$$

$$S.G. = \frac{\Sigma \text{ Pounds}_N}{\sum \frac{\text{Pounds}_N}{(S.G.)_N}}$$

$$\text{Pounds}_N = L_N \times (MW)_N$$

$$S.G. = \frac{\Sigma[L_N \times (MW)_N]}{\sum \left[\frac{L_N \times (MW)_N}{(S.G.)_N}\right]}$$

Column 5 lists a specific gravity for each component in the liquid phase at standard conditions except as noted. It would be more accurate to adjust these gravities for the actual pressure and temperature of the fluid being flashed. This will have a marginal effect on the results as the characterization of the heptanes is of overriding importance to the calculation. If this is not known for the pressure and temperature of the flash it can be approximated from known conditions using Figure 3-8.

Column 6 is derived by dividing Column 4 by Column 5. The liquid specific gravity is:

$$S.G. = \frac{7{,}212}{9{,}238} = 0.78$$

The specific gravity in °API can be calculated from Equation 3-1 as:

$$°API = \frac{141.5}{0.78} - 131.5 = 49.9$$

Liquid Flow Rate

The liquid flow rate in moles per day for a given inlet stream can be determined from Equation 3-17. In our example, for an inlet stream rate of 10,000 moles per day, the liquid flow rate is:

$$L = \frac{10,000}{1 + \frac{50.1}{49.9}} = 4,990 \text{ moles per day}$$

The liquid flow rate in barrels per day can be derived from:

$$Q_l = \frac{L \times (\overline{MW})}{350\,(S.G.)} \qquad (3-17)$$

where Q_l = liquid flow rate, bpd
S.G. = specific gravity of liquid (water = 1)

Derivation of Equation 3-17

There are 350 pounds per barrel of water, and 350 (S.G.) pounds per barrel of liquid.

Pounds of liquid = $L \times (\overline{MW})$

$$Q_l = \frac{\text{Pounds}}{\text{Pounds per Barrel}}$$

$$Q_l = \frac{L \times (MW)}{350\,(S.G.)}$$

For our example:

$$Q_l = \frac{(4,990)\,(145)}{(350)\,(0.78)} = 2,650 \text{ bpd}$$

The Flow Stream

Many times the designer is given the mole fraction of each component in the feed stream but is not given the mole flow rate for the stream. It may be necessary to estimate the total number of moles in the feed stream (F) from an expected stock tank oil flow rate. As a first approximation, it can be assumed that all the oil in the stock tank can be characterized by the C_7^+ component of the stream. Thus, the feed rate in moles per day can be approximated as:

$$L \cong \frac{350\,(S.G.)_7\,Q_l}{(MW)_7} \tag{3-18}$$

where L = liquid flow rate, moles per day
$(S.G.)_7$ = specific gravity of C_7^+
$(MW)_7$ = molecular weight of C_7^+
Q_l = flow rate of liquid, bpd

The mole flow rate of the feed stream is then calculated as:

$$F = \frac{L}{(\text{Mole Fraction})_7} \tag{3-19}$$

where F = flow rate of feed stream, moles per day
$(\text{Mole Fraction})_7$ = mole fraction of the C_7^+ component in the feed stream

For our example, if the mole feed rate of 10,000 moles per day was not given, but was required to design for 2,500 bpd of stock tank liquid, the mole feed rate for flash calculations would be approximated as:

$$L = \frac{350\,(0.86)\,(2,500)}{253} = 2,974$$

$$F = \frac{2,974}{0.2607} = 11,400 \text{ moles / day}$$

The flash calculation could then proceed. The calculated flow rates for each stream in the process could then be ratioed to reflect the error between assumed stock tank flow rate and desired stock tank flow rate.

APPROXIMATE FLASH CALCULATIONS

Most often, flash calculations are too involved and subject to too many arithmetic mistakes to be done by hand as previously detailed. They are usually done either on computer or in a programmable calculator. Sometimes it is necessary to get a quick estimate of the volume of gas that is expected to be flashed from an oil stream at various pressures.

Figure 3-27 was developed by flashing several crude oils of different gravities at different pressure ranges. The curves are approximate. The actual shape would depend on the initial separation pressure, the number and pressure of intermediate flashes, and the temperature.

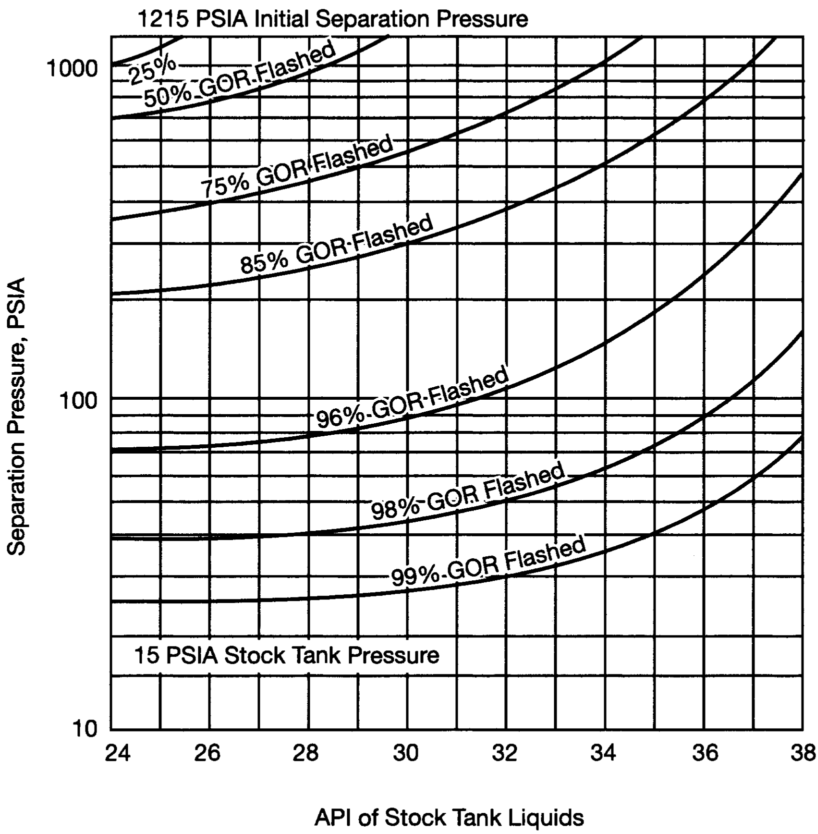

Figure 3-27. Preliminary estimation of % GOR flashed for given API of stock tank liquids and separation pressures—Gulf Coast crudes.

Use of the curve is best explained by an example. Suppose a 30°API crude with a GOR of 500 is flashed at 1,000 psia, 500 psia, and 50 psia before going to a stock tank. Roughly 50% of the gas that will eventually be flashed from the crude, or 250 ft^3/B, will be liberated as gas in the 1,000 psia separator. Another 25% (75%–50%), or 125 ft^3/B will be separated at 500 psia and 23% (98%–75%), or 115 ft^3/B will be separated at 50 psia. The remaining 10 ft^3/B (100%–98%) will be vented from the stock tank.

It must be stressed that Figure 3-27 is only to be used where a quick approximation, which could be subject to error, is acceptable. It cannot be used for estimating gas flashed from condensate produced in gas wells.

OTHER PROPERTIES

The iterative flash calculation detailed in Figure 3-27 shows one of many methods for calculating equilibrium conditions. Flash calculations are inherently rigorous and best performed by sophisticated simulation software, such as HYSIM or ASPEN. For preliminary considerations, however, correlations and simplified methods for flash calculations such as that detailed earlier may be used.

Once the equilibrium conditions (and, therefore, the gas and liquid compositions) are known, several very useful physical properties are obtainable, such as the dew point, the bubble point, the heating value (net and gross), and k, the ratio of gas specific heats.

Dew point–The point at which liquid first appears within a gas sample.

Bubble point–The point at which gas first appears within a liquid sample.

Net heating value–Heat released by combustion of a gas sample with water vapor as a combustion product; also known as the lower heating value (LHV).

Gross heating value–Heat released by combustion of a gas sample with liquid water as a combustion product; also known as the higher heating value (HHV).

k–Ratio of heat capacity at constant pressure (C_p) to heat capacity at constant volume (C_v). Often used in compressor calculations of horsepower requirements and volumetric efficiencies. This ratio is relatively constant for natural gas molecular weights and ranges between 1.2 and 1.3. See Figure 3-28.

A more precise definition of the dew point makes a distinction between the hydrocarbon dew point, representing the deposition of a hydrocarbon liquid, and the water dew point, representing the deposition of liquid water. Often, sales gas contracts specify control of the water dew point for hydrate and corrosion control and not the hydrocarbon dew point. In such cases, hydrocarbons will often condense in the pipeline as the gas cools (assuming that separation has occurred at a higher temperature than ambient), and provisions to separate this "condensate" must be provided.

The bubble point can also be referred to as the "true vapor pressure." A critical distinction lies here between the true vapor pressure and the Reid Vapor Pressure (RVP). The Reid Vapor Pressure is measured according to a specific ASTM standard (D323) and lies below the true vapor pressure.

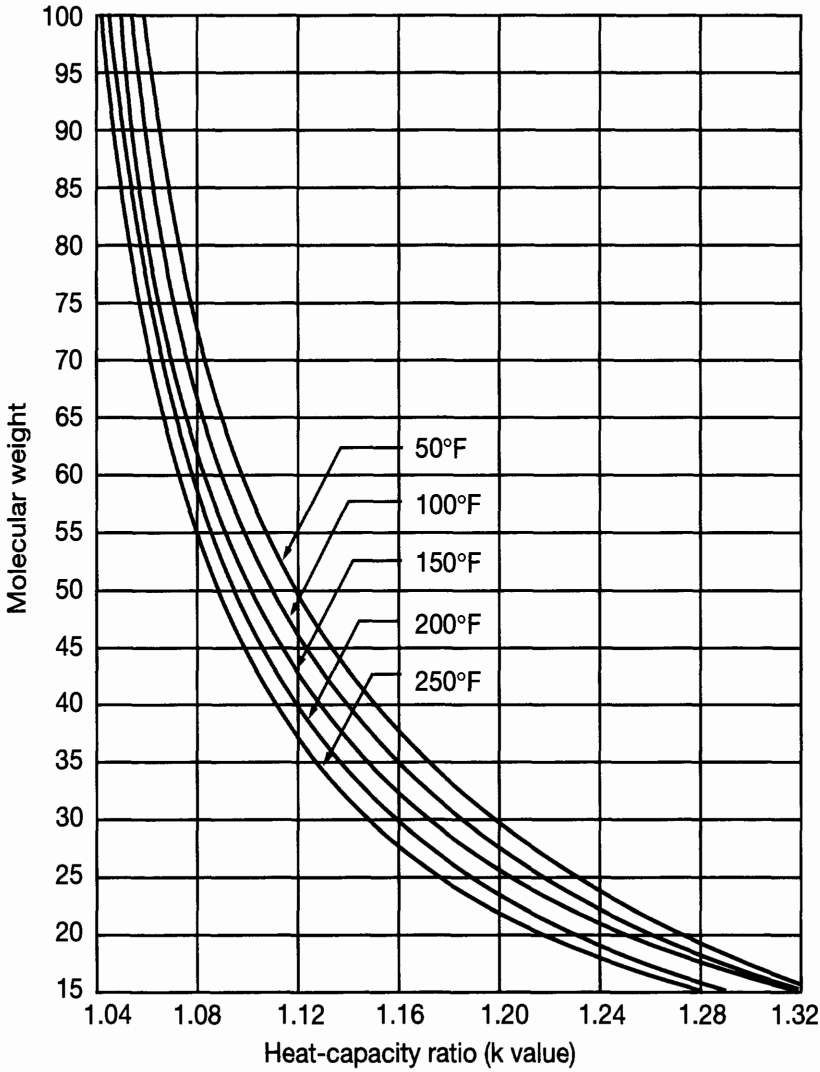

Figure 3-28. Approximate heat-capacity ratios of hydrocarbon gases (*courtesy of GPSA Engineering Data Book*).

The relationship between the two pressures is shown in Figure 3-29. (Note that an RVP below atmospheric pressure does not indicate that vapors will be absent from a sample at atmospheric pressure.)

(text continued on page 100)

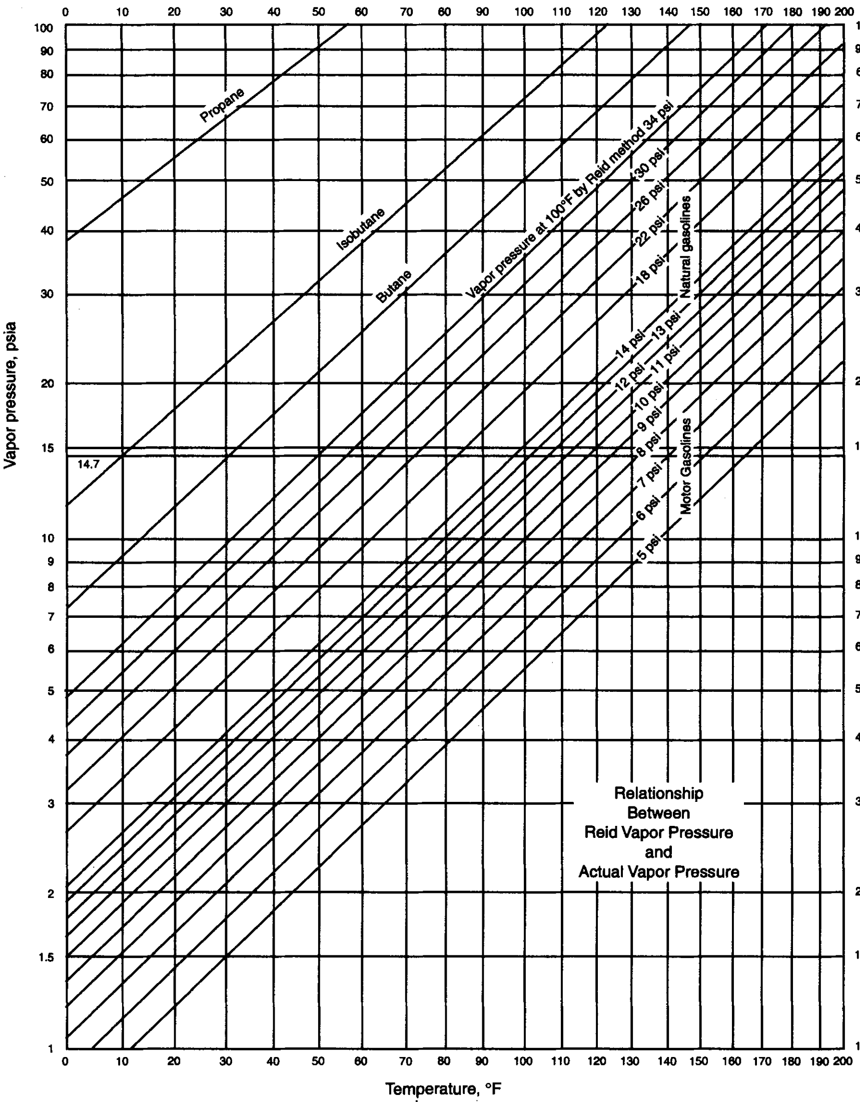

Figure 3-29. Relationship between Reid Vapor Pressure and actual vapor pressure (*courtesy of GPSA Engineering Data Book*).

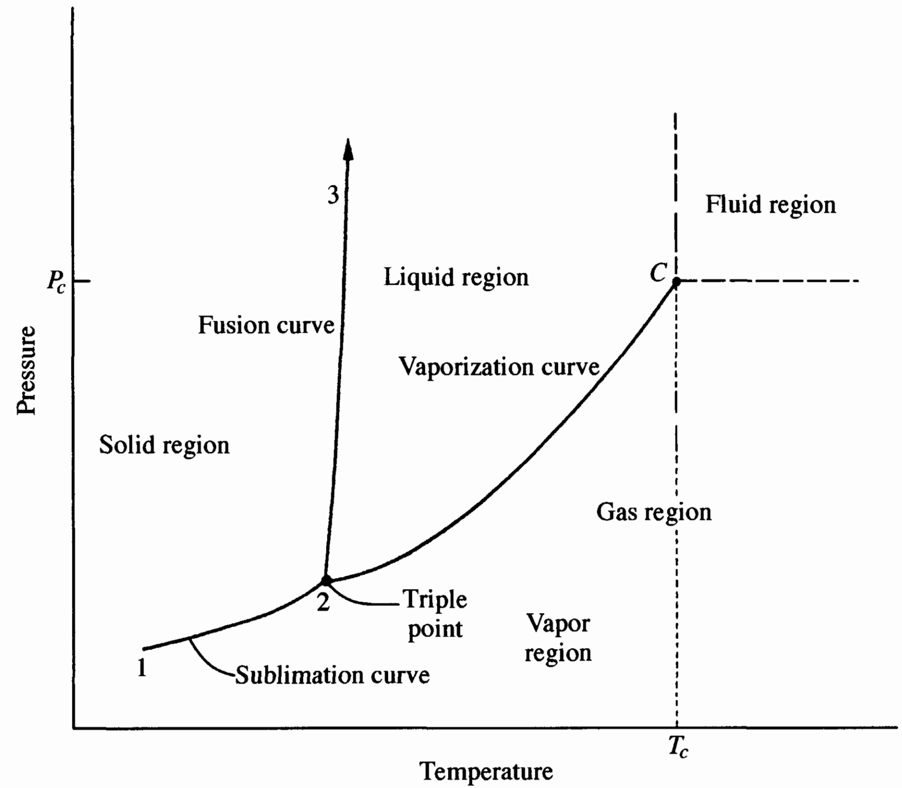

Figure 3-30. PT diagram for a pure substance (from *Introduction to Chemical Engineering Thermodynamics*, 4th Edition by J. M. Smith and H. C. Van Ness. New York: McGraw-Hill, Inc., 1987).

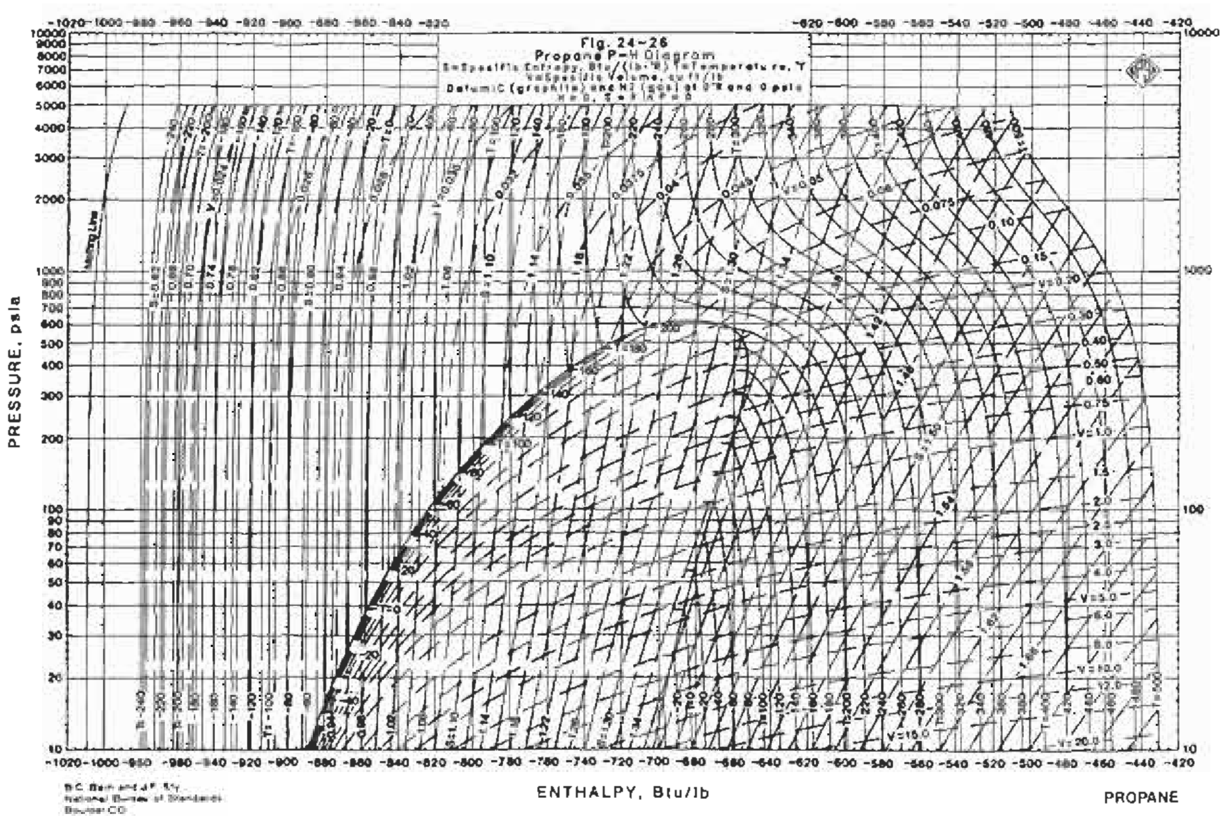

Figure 3-31. A P-H diagram for propane (*courtesy of GPSA Engineering Data Book*).

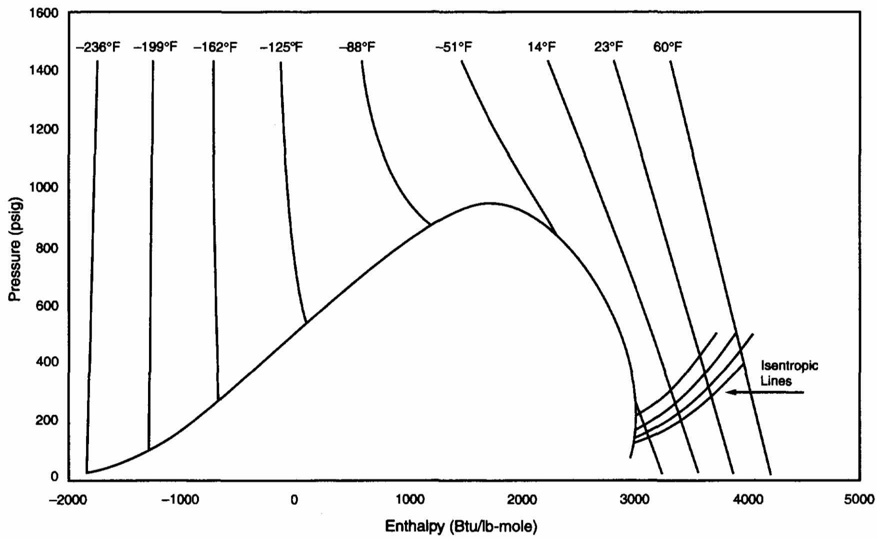

Figure 3-32. P-H diagram for 0.6 SG natural gas.

(text continued from page 97)

The bubble point and the dew point are equivalent for a single-component mixture only. A graphic representation of the bubble/dew point for a pure substance is shown as the vaporization curve in Figure 3-30.

A basic representation of the equilibrium information for a specific fluid composition can be found in a P-H (Pressure-Enthalpy) diagram, which is highly dependent on the sample composition. This diagram can be used to investigate thermodynamic fluid properties as well as thermodynamic phenomena such as retrograde condensation and the Joule-Thomson effect. Please note, however, that a P-H diagram is unlikely to be available for anything but a single component of the mixture, unless the diagram is created by simulation software packages such as those mentioned above. A P-H diagram for propane is shown in Figure 3-31; a P-H diagram for a 0.6 specific gravity natural gas is shown in Figure 3-32.

CHAPTER

4

Two-Phase Oil and Gas Separation*

INTRODUCTION

Produced wellhead fluids are complex mixtures of different compounds of hydrogen and carbon, all with different densities, vapor pressures, and other physical characteristics. As a well stream flows from the hot, high-pressure petroleum reservoir, it experiences pressure and temperature reductions. Gases evolve from the liquids and the well stream changes in character. The velocity of the gas carries liquid droplets, and the liquid carries gas bubbles. The physical separation of these phases is one of the basic operations in the production, processing, and treatment of oil and gas.

In oil and gas separator design, we mechanically separate from a hydrocarbon stream the liquid and gas components that exist at a specific temperature and pressure. Proper separator design is important because a separation vessel is normally the initial processing vessel in any facility, and improper design of this process component can "bottleneck" and reduce the capacity of the entire facility.

*Reviewed for the 1998 edition by Mary E. Thro of Paragon Engineering Services, Inc.

Separators are classified as "two-phase" if they separate gas from the total liquid stream and "three-phase" if they also separate the liquid stream into its crude oil and water components. This chapter deals with two-phase separators. In addition, it discusses the requirements of good separation design and how various mechanical devices take advantage of the physical forces in the produced stream to achieve good separation.

Separators are sometimes called "gas scrubbers" when the ratio of gas rate to liquid rate is very high. Some operators use the term "traps" to designate separators that handle flow directly from wells. In any case, they all have the same configuration and are sized in accordance with the same procedure.

FACTORS AFFECTING SEPARATION

Characteristics of the flow stream will greatly affect the design and operation of a separator. The following factors must be determined before separator design:

- Gas and liquid flow rates (minimum, average, and peak)
- Operating and design pressures and temperatures
- Surging or slugging tendencies of the feed streams
- Physical properties of the fluids such as density and compressibility
- Designed degree of separation (e.g., removing 100% of particles greater than 10 microns)
- Presence of impurities (paraffin, sand, scale, etc.)
- Foaming tendencies of the crude oil
- Corrosive tendencies of the liquids or gas

EQUIPMENT DESCRIPTION

Horizontal Separators

Separators are designed in either horizontal, vertical, or spherical configurations. Figure 4-1 is a schematic of a horizontal separator. The fluid enters the separator and hits an inlet diverter causing a sudden change in momentum. The initial gross separation of liquid and vapor occurs at the inlet diverter. The force of gravity causes the liquid droplets to fall out of the gas stream to the bottom of the vessel where it is collected. This liq-

uid collection section provides the retention time required to let entrained gas evolve out of the oil and rise to the vapor space. It also provides a surge volume, if necessary, to handle intermittent slugs of liquid. The liquid then leaves the vessel through the liquid dump valve. The liquid dump valve is regulated by a level controller. The level controller senses changes in liquid level and controls the dump valve accordingly.

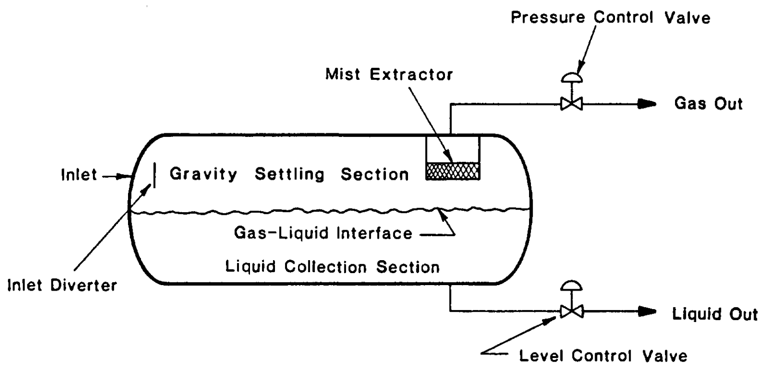

Figure 4-1. Horizontal separator schematic.

The gas flows over the inlet diverter and then horizontally through the gravity settling section above the liquid. As the gas flows through this section, small drops of liquid that were entrained in the gas and not separated by the inlet diverter are separated out by gravity and fall to the gas-liquid interface.

Some of the drops are of such a small diameter that they are not easily separated in the gravity settling section. Before the gas leaves the vessel it passes through a coalescing section or mist extractor. This section uses elements of vanes, wire mesh, or plates to coalesce and remove the very small droplets of liquid in one final separation before the gas leaves the vessel.

The pressure in the separator is maintained by a pressure controller. The pressure controller senses changes in the pressure in the separator and sends a signal to either open or close the pressure control valve accordingly. By controlling the rate at which gas leaves the vapor space of the vessel the pressure in the vessel is maintained. Normally, horizontal separators are operated half full of liquid to maximize the surface area of the gas liquid interface.

Vertical Separators

Figure 4-2 is a schematic of a vertical separator. In this configuration the inlet flow enters the vessel through the side. As in the horizontal separator, the inlet diverter does the initial gross separation. The liquid flows down to the liquid collection section of the vessel. Liquid continues to flow downward through this section to the liquid outlet. As the liquid reaches equilibrium, gas bubbles flow counter to the direction of the liquid flow and eventually migrate to the vapor space. The level controller and liquid dump valve operate the same as in a horizontal separator.

The gas flows over the inlet diverter and then vertically upward toward the gas outlet. In the gravity settling section the liquid drops fall vertically downward counter to the gas flow. Gas goes through the mist extractor section before it leaves the vessel. Pressure and level are maintained as in a horizontal separator.

Spherical Separators

A typical spherical separator is shown in Figure 4-3. The same four sections can be found in this vessel. Spherical separators are a special

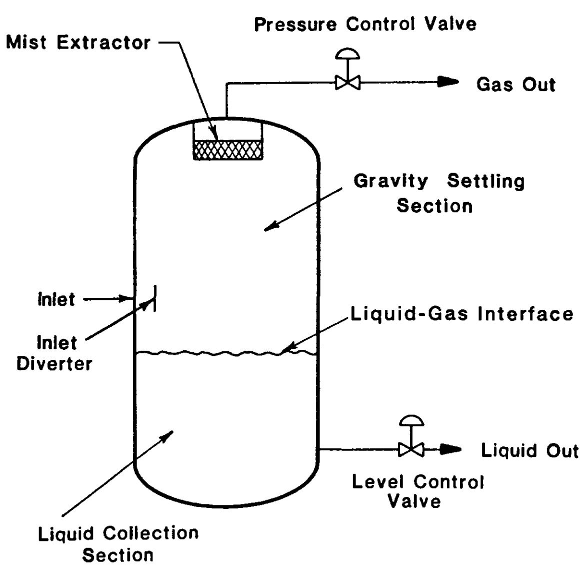

Figure 4-2. Vertical separator schematic.

case of a vertical separator where there is no cylindrical shell between the two heads. They may be very efficient from a pressure containment standpoint but because (1) they have limited liquid surge capability and (2) they exhibit fabrication difficulties, they are not usually used in oil field facilities. For this reason we will not be discussing spherical separators any further.

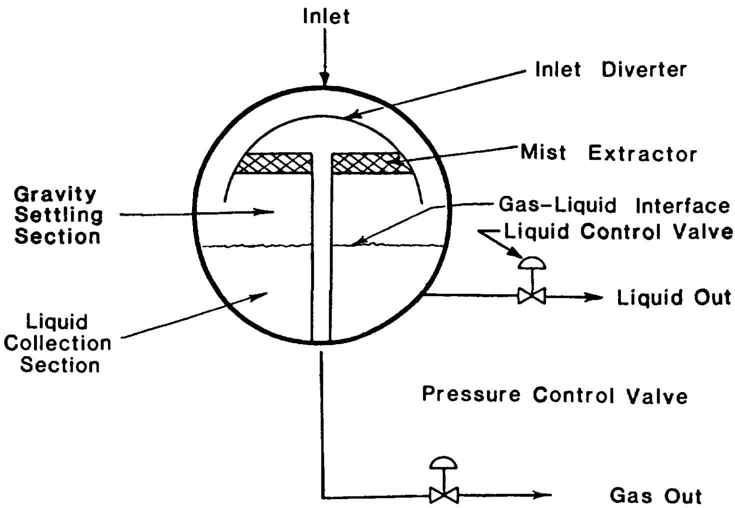

Figure 4-3. Spherical separator schematic.

Other Configurations

Cyclone separators are designed to operate by centrifugal force. These designs are best suited for fairly clean gas streams. The swirling action of the gas stream as it enters the scrubber separates the droplets and dust from the gas stream by centrifugal force. Although such designs can result in significantly smaller sizes, they are not commonly used in production operations because (1) their design is rather sensitive to flow rate and (2) they require greater pressure drop than the standard configurations previously described. Since separation efficiency decreases as velocity decreases, cyclone separators are not suitable for widely varying flow rates. These units are commonly used to recover glycol carryover downstream of a dehydration tower. In recent years, demand for using cyclone separators on floating facilities has increased because space and weight considerations are overriding on such facilities.

Two-barrel separators are common where there is a very low liquid flow rate. In these designs the gas and liquid chambers are separated as shown in Figure 4-4. The flow stream enters the vessel in the upper barrel and strikes the inlet diverter. The free liquids fall to the lower barrel through a flow pipe. The gas flows through the gravity settling section and encounters a mist extractor en route to the gas outlet. The liquids drain through a flow pipe into the lower barrel. Small amounts of gas entrained in the liquid are liberated in the liquid collection barrel and flow up through the flow pipes. In this manner the liquid accumulation is separated from the gas stream so that there is no chance of high gas velocities re-entraining liquid as it flows over the interface. Because of their additional cost, and the absence of problems with single vessel separators, they are not widely used in oil field systems.

Another type of separator that is frequently used in some high-gas/low-liquid flow applications is a filter separator. These can be either horizontal or vertical in configuration. Figure 4-5 shows a horizontal two-barrel design. Filter tubes in the initial separation section cause coalescence of any liquid mist into larger droplets as the gas passes through the tubes. A secondary section of vanes or other mist extractor elements

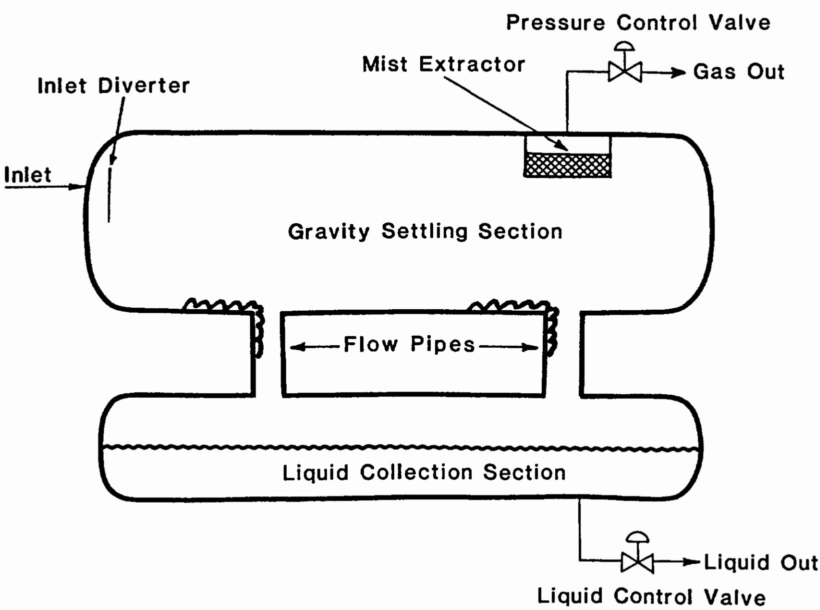

Figure 4-4. Double-barrel separator.

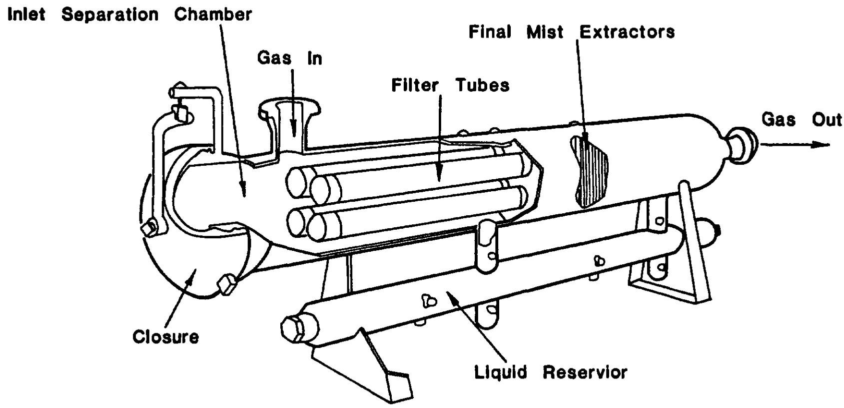

Figure 4-5. Typical filter separator.

removes these coalesced droplets. This vessel can remove 100% of all particles larger than about 2 microns and 99% of those down to about ½ micron. Filter separators are commonly used on compressor inlets in field compressor stations, final scrubbers upstream of glycol contact towers, and instrument/fuel gas applications. The design of filter separators is proprietary and dependent upon the type of filter element employed.

In applications where there is very little liquid flow, often a horizontal separator will be designed with a liquid sump on the outlet end to provide the required liquid retention time. This results in an overall smaller diameter for the vessel.

Scrubbers

A scrubber is a two-phase separator that is designed to recover liquids carried over from the gas outlets of production separators or to catch liquids condensed due to cooling or pressure drops. Liquid loading in a scrubber is much lower than that in a separator. Typical applications include: upstream of mechanical equipment such as compressors that could be damaged, destroyed or rendered ineffective by free liquid; downstream of equipment that can cause liquids to condense from a gas stream (such as coolers); upstream of gas dehydration equipment that would lose efficiency, be damaged, or be destroyed if contaminated with liquid hydrocarbons; and upstream of a vent or flare outlet.

Vertical scrubbers are most commonly used. Horizontal scrubbers can be used, but space limitations usually dictate the use of a vertical configuration.

HORIZONTAL VS. VERTICAL VESSEL SELECTION

Horizontal separators are smaller and less expensive than vertical separators for a given gas capacity. In the gravity settling section of a horizontal vessel, the liquid droplets fall perpendicular to the gas flow and thus are more easily settled out of the gas continuous phase. Also, since the interface area is larger in a horizontal separator than a vertical separator, it is easier for the gas bubbles, which come out of solution as the liquid approaches equilibrium, to reach the vapor space. Horizontal separators offer greater liquid capacity and are best suited for liquid-liquid separation and foaming crudes.

Thus, from a pure gas/liquid separation process, horizontal separators would be preferred. However, they do have the following drawbacks, which could lead to a preference for a vertical separator in certain situations:

1. Horizontal separators are not as good as vertical separators in handling solids. The liquid dump of a vertical separator can be placed at the center of the bottom head so that solids will not build up in the separator but continue to the next vessel in the process. As an alternative, a drain could be placed at this location so that solids could be disposed of periodically while liquid leaves the vessel at a slightly higher elevation.

   In a horizontal vessel, it is necessary to place several drains along the length of the vessel. Since the solids will have an angle of repose of 45° to 60°, the drains must be spaced at very close intervals. Attempts to lengthen the distance between drains, by providing sand jets in the vicinity of each drain to fluidize the solids while the drains are in operation, are expensive and have been only marginally successful in field operations.

2. Horizontal vessels require more plan area to perform the same separation as vertical vessels. While this may not be of importance at a land location, it could be very important offshore.

3. Smaller, horizontal vessels can have less liquid surge capacity than vertical vessels sized for the same steady-state flow rate. For a given change in liquid surface elevation, there is typically a larger increase in liquid volume for a horizontal separator than for a vertical separator sized for the same flow rate. However, the geometry of a horizontal vessel causes any high level shut-down device to be located

close to the normal operating level. In a vertical vessel the shutdown could be placed much higher, allowing the level controller and dump valve more time to react to the surge. In addition, surges in horizontal vessels could create internal waves that could activate a high level sensor.

It should be pointed out that vertical vessels also have some drawbacks that are not process related and must be considered in making a selection. These are:

1. The relief valve and some of the controls may be difficult to service without special ladders and platforms.
2. The vessel may have to be removed from a skid for trucking due to height restrictions.

Overall, horizontal vessels are the most economical for normal oil-gas separation, particularly where there may be problems with emulsions, foam, or high gas-oil ratios. Vertical vessels work most effectively in low GOR applications. They are also used in some very high GOR applications, such as scrubbers where only fluid mists are being removed from the gas.

VESSEL INTERNALS

Inlet Diverters

There are many types of inlet diverters. Two main types are baffle plates (shown in Figure 4-6) and centrifugal diverters (shown in Figure 4-7). A baffle plate can be a spherical dish, flat plate, angle iron, cone, or just about anything that will accomplish a rapid change in direction and velocity of the fluids and thus disengage the gas and liquid. The design of

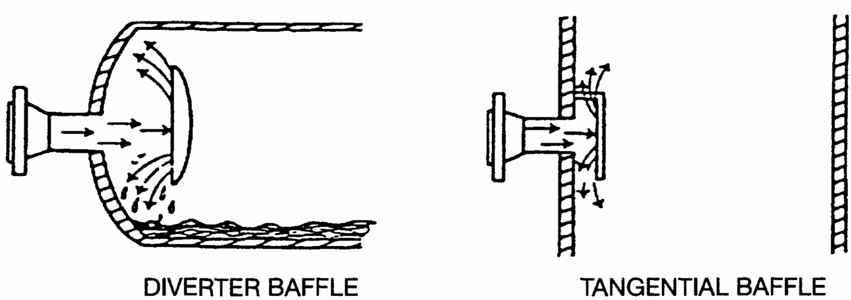

Figure 4-6. Baffle plates.

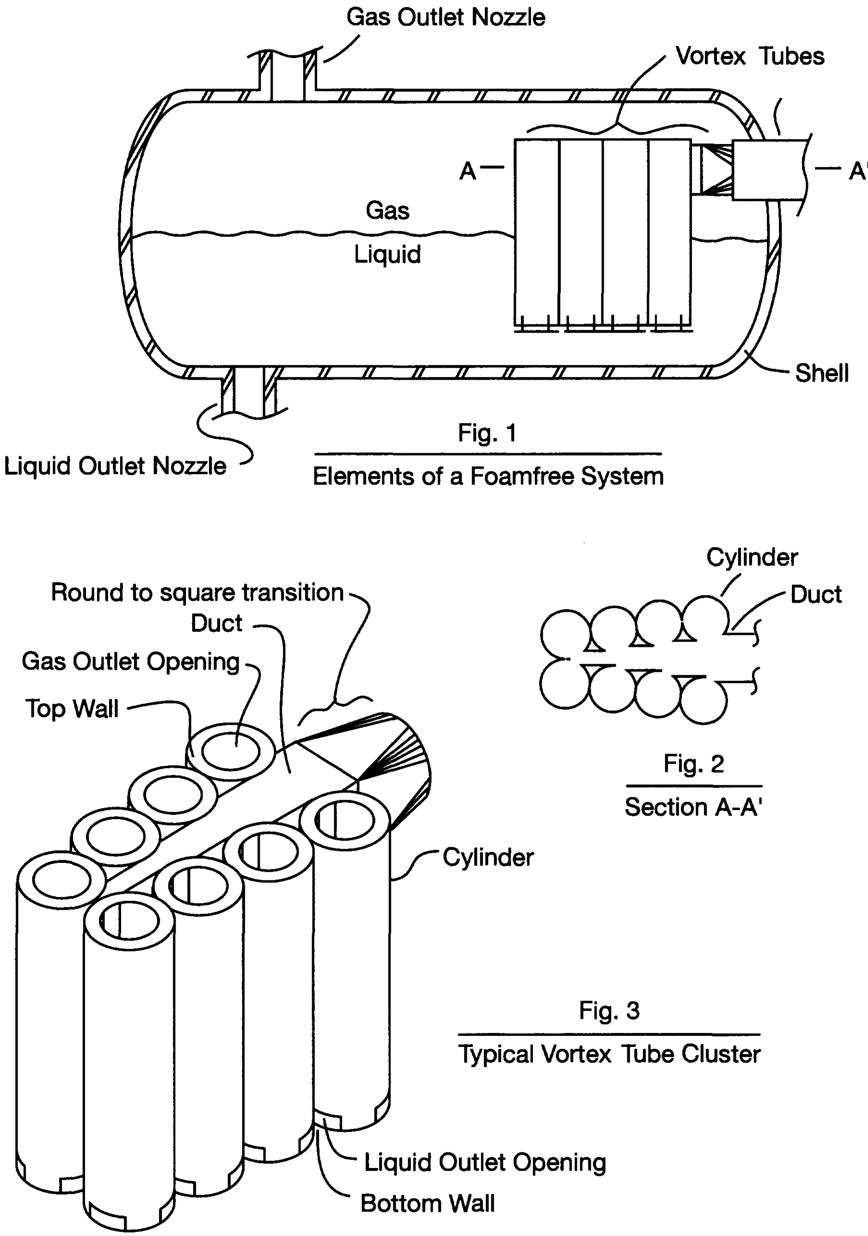

Figure 4-7. Three views of an example centrifugal inlet diverter (*courtesy of Porta-Test Systems, Inc.*).

the baffles is governed principally by the structural supports required to resist the impact-momentum load. The advantage of using devices such as a half sphere or cone is that they create less disturbance than plates or angle iron, cutting down on re-entrainment or emulsifying problems.

Centrifugal inlet diverters use centrifugal force, rather than mechanical agitation, to disengage the oil and gas. These devices can have a cyclonic chimney or may use a tangential fluid race around the walls. Centrifugal inlet diverters are proprietary but generally use an inlet nozzle sufficient to create a fluid velocity of about 20 fps. Centrifugal diverters work well in initial gas separation and help to prevent foaming in crudes.

Wave Breakers

In long horizontal vessels it is necessary to install wave breakers, which are nothing more than vertical baffles spanning the gas-liquid interface and perpendicular to the flow.

Defoaming Plates

Foam at the interface may occur when gas bubbles are liberated from the liquid. This foam can be stabilized with the addition of chemicals at the inlet. Many times a more effective solution is to force the foam to pass through a series of inclined parallel plates or tubes as shown in Figure 4-8 so as to aid in coalescence of the foam bubbles.

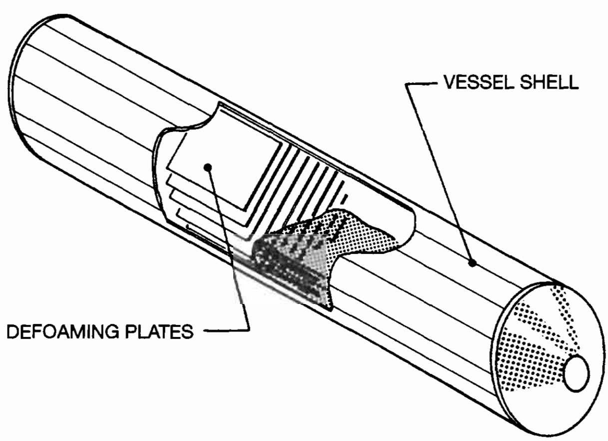

Figure 4-8. Defoaming plates.

Vortex Breaker

It is normally a good idea to include a simple vortex breaker as shown in Figure 4-9 to keep a vortex from developing when the liquid control valve is open. A vortex could suck some gas out of the vapor space and re-entrain it in the liquid outlet.

Mist Extractor

Mist extractors can be made of wire mesh, vanes, centrifugal force devices, or packing. Wire mesh pads (Figure 4-10) are made of finely woven mats of stainless steel wire wrapped into a tightly packed cylinder. The liquid droplets impinge on the matted wires and coalesce. The effectiveness of wire mesh depends largely on the gas being in the proper velocity range. If the velocities are too high, the liquids knocked out will be re-entrained. If the velocities are low, the vapor just drifts through the mesh element without the droplets impinging and coalescing.

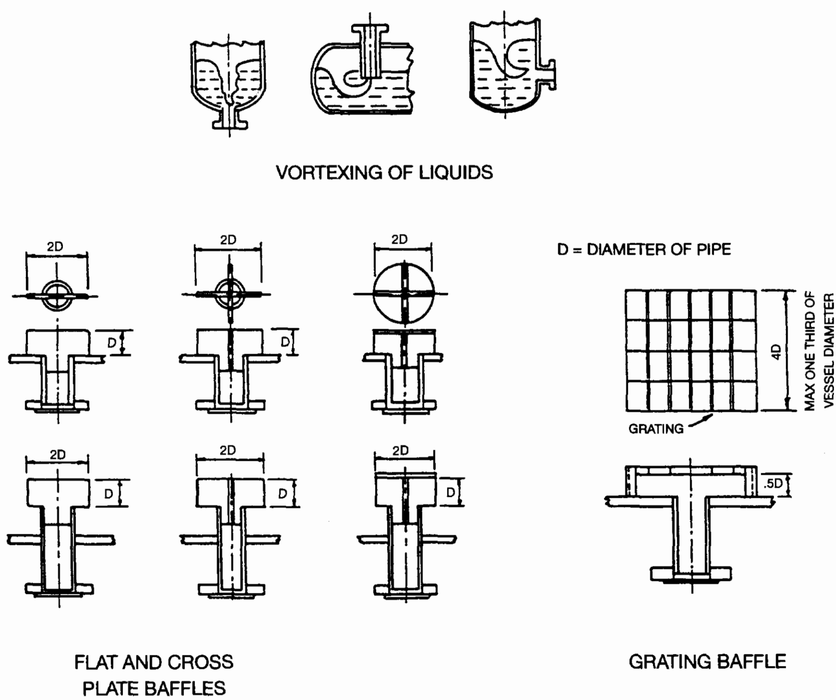

Figure 4-9. Typical vortex breakers.

Figure 4-10. Example wire mesh mist eliminator (*photo courtesy of ACS Industries, LP, Houston, Texas*).

The construction is often specified by calling for a certain thickness (usually 3 to 7 inches) and mesh density (usually 10 to 12 pounds per cubic foot). Experience has indicated that a properly sized wire mesh eliminator can remove 99% of 10-micron and larger droplets. Although wire mesh eliminators are inexpensive they are more easily plugged than the other types.

Vane eliminators (Figure 4-11) force the gas flow to be laminar between parallel plates that contain directional changes. Figure 4-12 shows a vane mist extractor made from angle iron. In vane eliminators, droplets impinge on the plate surface where they coalesce and fall to a liquid collecting spot. They are routed to the liquid collection section of the vessel. Vane-type eliminators are sized by their manufacturers to assure both laminar flow and a certain minimum pressure drop.

Some separators have centrifugal mist eliminators (as shown in Figure 4-13) that cause the liquid drops to be separated by centrifugal force. These can be more efficient than either wire mesh or vanes and are the least susceptible to plugging. However, they are not in common use in production operations because their removal efficiencies are sensitive to small changes in flow. In addition, they require relatively large pressure drops to create the centrifugal force. To a lesser extent, random packing is sometimes used for mist extraction, as shown in Figure 4-14. The packing acts as a coalescer.

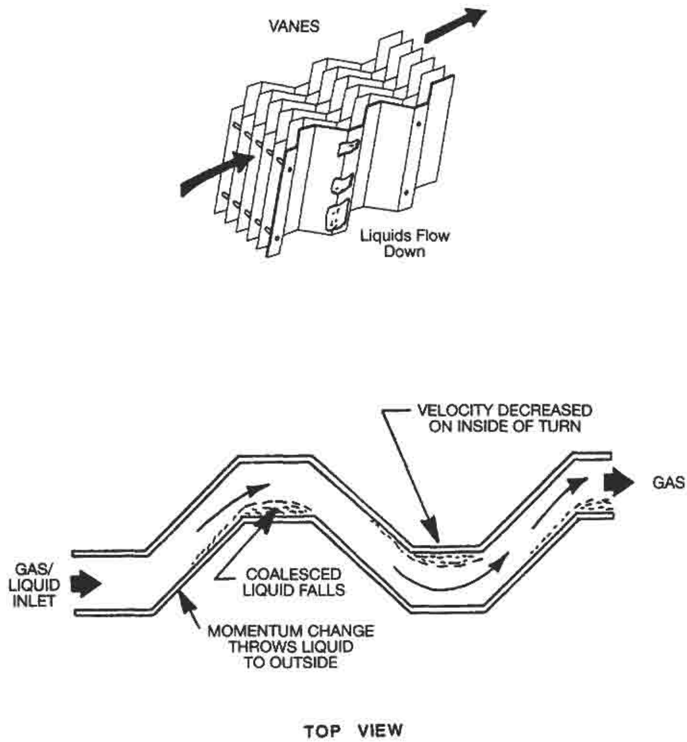

Figure 4-11. Typical mist extractor.

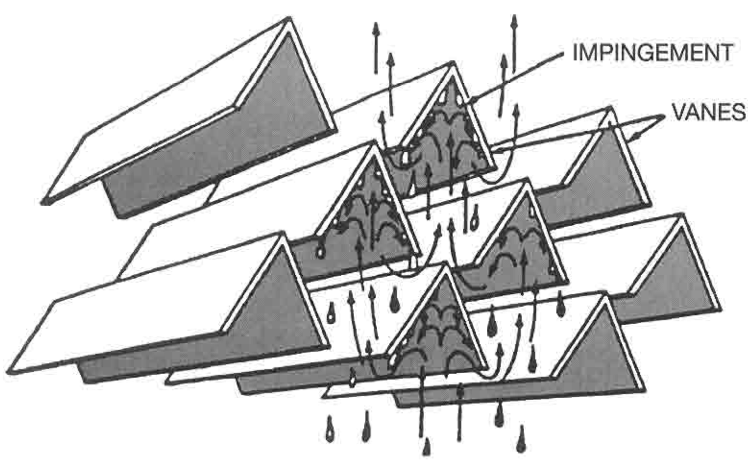

Figure 4-12. A vane mist extractor made from angle iron.

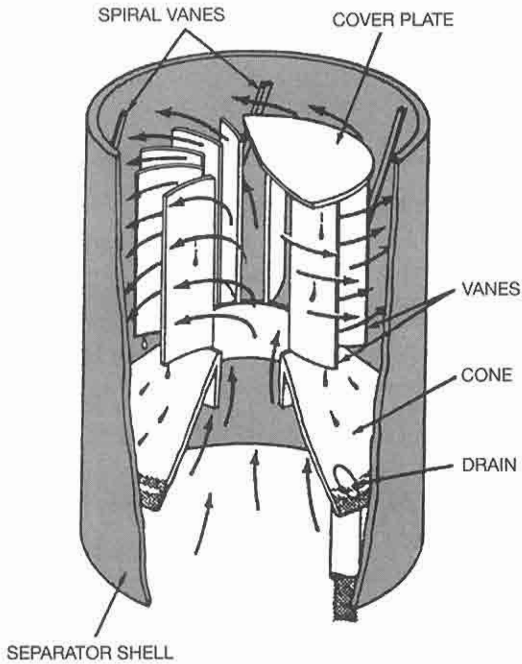

Figure 4-13. A centrifugal mist eliminator.

Figure 4-14. A coalescing pack mist extractor.

POTENTIAL OPERATING PROBLEMS

Foamy Crudes

The major cause of foam in crude oil is the appearance of impurities, other than water, which are impractical to remove before the stream reaches the separator. Foam presents no problem within a separator if the

internal design assures adequate time or sufficient coalescing surface for the foam to "break."

Foaming in a separating vessel is a threefold problem:

1. Mechanical control of liquid level is aggravated because any control device must deal with essentially three liquid phases instead of two.
2. Foam has a large volume-to-weight ratio. Therefore, it can occupy much of the vessel space that would otherwise be available in the liquid collecting or gravity settling sections.
3. In an uncontrolled foam bank, it becomes impossible to remove separated gas or degassed oil from the vessel without entraining some of the foamy material in either the liquid or gas outlets.

Comparison of foaming tendencies of a known oil to a new one, about which no operational information is known, provides an understanding of the relative foam problem that may be expected with the new oil as weighed against the known oil. A related amount of adjustment can then be made in the design parameters, as compared to those found satisfactory for the known case.

It should be noted that the amount of foam is dependent on the pressure drop to which the inlet liquid is subjected, as well as the characteristics of the liquid at separator conditions. In some cases, the effect of temperature may be significant.

Foam depressants often will do a good job in increasing the capacity of a given separator. However, in sizing a separator to handle a particular crude, the use of an effective depressant should not be assumed because characteristics of the crude and of the foam may change during the life of the field. Also, the cost of foam depressants for high rate production may be prohibitive. Sufficient capacity should be provided in the separator to handle the anticipated production without use of a foam depressant or inhibitor. Once placed in operation, a foam depressant may allow more throughput than the design capacity.

Paraffin

Separator operation can be adversely affected by an accumulation of paraffin. Coalescing plates in the liquid section and mesh pad mist extractors in the gas section are particularly prone to plugging by accumulations of paraffin. Where it is determined that paraffin is an actual or potential problem, the use of plate-type or centrifugal mist extractors

should be considered. Manways, handholes, and nozzles should be provided to allow steam, solvent, or other types of cleaning of the separator internals. The bulk temperature of the liquid should always be kept above the cloud point of the crude oil.

Sand

Sand can be very troublesome in separators by causing cutout of valve trim, plugging of separator internals, and accumulation in the bottom of the separator. Special hard trim can minimize the effects of sand on the valves. Accumulations of sand can be alleviated by the use of sand jets and drains.

Plugging of the separator internals is a problem that must be considered in the design of the separator. A design that will promote good separation and have a minimum of traps for sand accumulation may be difficult to attain, since the design that provides the best mechanism for separating the gas, oil, and water phases probably will also provide areas for sand accumulation. A practical balance for these factors is the best solution.

Liquid Carryover and Gas Blowby

Liquid carryover and gas blowby are two common operating problems. Liquid carryover occurs when free liquid escapes with the gas phase and can indicate high liquid level, damage to vessel internals, foam, improper design, plugged liquid outlets, or a flow rate that exceeds the vessel's design rate.

Gas blowby occurs when free gas escapes with the liquid phase and can be an indication of low liquid level, vortexing, or level control failure.

THEORY

Settling

In the gravity settling section the liquid drops will settle at a velocity determined by equating the gravity force on the drop with the drag force caused by its motion relative to the gas continuous phase.

The drag force is determined from the equation:

$$F_D = C_D A\rho \left[\frac{V_t^2}{2g}\right] \tag{4-1}$$

where F_D = drag force, lb
C_D = drag coefficient
A = cross-sectional area of the droplet, ft^2
ρ = density of the continuous phase, lb/ft^3
V_t = terminal settling velocity of the droplet, ft/s
g = gravitational constant, 32.2 ft/s^2

If the flow around the drop were laminar, then Stokes' Law would govern and:

$$C_D = \frac{24}{Re} \tag{4-2}$$

where Re = Reynolds number

It can be shown that in such a gas the droplet settling velocity would be given by:

$$V_t = \frac{1.78 \times 10^{-6} \, (\Delta S.G.) \, d_m^2}{\mu} \tag{4-3}$$

where $\Delta S.G.$ = difference in specific gravity relative to water of the drop and the gas
d_m = drop diameter, micron
μ = viscosity of the gas, cp

Derivation of Equation 3

For low Reynolds number flows, i.e., Re < 1

$$C_D = \frac{24}{Re}$$

The drag force is then

$$F_D = C_D \, A\rho_g \frac{V^2}{2g} = \frac{24}{Re}\left(\pi \frac{D^2}{4}\right)\rho_g \frac{V^2}{2g}$$

$$= \frac{24}{\dfrac{\rho_g}{g}\dfrac{Dv}{\mu'}}\left(\pi \frac{D^2}{4}\right)\rho_g \frac{V^2}{2g}$$

D = drop diameter, ft
μ' = viscosity, lb-sec/ft^2
$F_D = 3\pi\mu' \, Dv$ (Stokes' Law)

The buoyant force on a sphere from Archimedes' principles is

$$F_B = (\rho_l - \rho_g)\frac{\pi D^3}{6}$$

When the drag force is equal to the buoyancy force, the droplet's acceleration is zero so that it moves at a constant velocity. This is the terminal velocity.

$$F_D = F_B$$

$$3\pi\mu' VD = (\rho_l - \rho_g)\frac{\pi D^3}{6}$$

$$V_t = \frac{(\rho_l - \rho_g)D^2}{18\mu'}$$

$$\mu' = \mu\,(2.088 \times 10^{-5})$$

where μ = viscosity, cp
$D = (d_m)\,(3.281 \times 10^{-6})$

where d_m = diameter, micron
$\rho_l = 62.4 \times S.G.$
$\rho_g = 62.4 \times S.G.$

where S.G. = specific gravity relative to water

$$V_t = \frac{62.4\,(\Delta S.G.)\,(3.281 \times 10^{-6} \times d_m)^2}{18\,(\mu)\,(2.088 \times 10^{-5})}$$

$$V_t = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)d_m^2}{\mu}$$

Unfortunately, for production facility design it can be shown that Stokes' Law does *not* govern, and the following more complete formula for drag coefficient must be used:

$$C_D = \frac{24}{Re} + \frac{3}{Re^{1/2}} + 0.34 \qquad (4-4)$$

Equating drag and buoyant forces, the terminal settling velocity is given by:

$$V_t = 0.0199\left[\left(\frac{\rho_l - \rho_g}{\rho_g}\right)\frac{d_m}{C_D}\right]^{1/2} \qquad (4-5)$$

where ρ_1 = density of liquid, lb/ft^3
ρ_g = density of the gas at the temperature and pressure in the separator, lb/ft^3

Derivation of Equation 4-5

C_D = constant

The drag force is then:

$$F_D = C_D A \rho_g \frac{V^2}{2g} = C_D \left(\frac{\pi D^2}{4}\right) \rho_g \frac{V^2}{2g}$$

when $F_B = F_D$,

$$C_D = \left(\frac{\pi D^2}{4}\right) \rho_g \frac{V^2}{2g} = (\rho_1 - \rho_g) \frac{\pi D^3}{6}$$

$$V_t = 6.55 \left[\left(\frac{\rho_1 - \rho_g}{\rho_g}\right) \frac{D}{C_D}\right]^{1/2}$$

$$D = d_m\ (3.281 \times 10^{-6})$$

$$V_t = 0.0199 \left[\left(\frac{\rho_1 - \rho_g}{\rho_g}\right) \frac{d_m}{C_D}\right]^{1/2}$$

For $C_D = 0.34$

$$V_t = 0.0204 \left[\left(\frac{\rho_1 - \rho_g}{\rho_g}\right) d_m\right]^{1/2}$$

Equations 4-4 and 4-5 can be solved by an iterative solution as follows:

1. Start with $V_t = 0.0204 \left[\dfrac{(\rho_1 - \rho_g)\, d_m}{\rho_g}\right]^{1/2}$

2. Calculate $\text{Re} = 0.0049 \dfrac{\rho_g d_m V}{\mu}$

3. From Re, calculate C_D using

$$C_D = \frac{24}{Re} + \frac{3}{Re^{1/2}} + 0.34$$

4. Recalculate V_t using

$$V_t = 0.0119\left[\left(\frac{\rho_l - \rho_g}{\rho_g}\right)\frac{d_m}{C_D}\right]^{1/2}$$

5. Go to step 2 and iterate.

Drop Size

The purpose of the gas separation section of the vessel is to condition the gas for final polishing by the mist extractor. From field experience it appears that if 100-micron drops are removed in this section, the mist extractor will not become flooded and will be able to perform its job of removing those drops between 10- and 100-micron diameter.

The gas capacity design equations in this section are all based on 100-micron removal. In some cases, this will give an overly conservative solution. The techniques used here can be easily modified for any drop size.

In this book we are addressing separators used in oil field facilities. These vessels usually require a gas separation section. There are special cases where the separator is designed to remove only very small quantities of liquid that could condense due to temperature or pressure changes in a stream of gas that has already passed through a separator and a mist extractor. These separators, commonly called "gas scrubbers," could be designed for removal of droplets on the order of 500 microns without fear of flooding their mist extractors. Fuel gas scrubbers, compressor suction scrubbers, and contact tower inlet scrubbers are examples of vessels to which this might apply.

Flare or vent scrubbers are designed to keep large slugs of liquid from entering the atmosphere through the vent or relief systems. In vent systems the gas is discharged directly to the atmosphere and it is common to design the scrubbers for removal of 300- to 500-micron droplets in the gravity settling section. A mist extractor is not included because of the possibility that it might plug creating a safety hazard. In flare systems, where the gas is discharged through a flame, there is the possibility that burning liquid droplets could fall to the ground before being consumed. It is still common to size the gravity settling section for 300- to 500-

micron removal, which the API guideline for refinery flares indicates is adequate to assure against a falling flame. In critical locations, such as offshore platforms, many operators include a mist extractor as an extra precaution against a falling flame. If a mist extractor is used, it is necessary to provide safety relief protection around the mist extractor in the event that it becomes plugged.

Retention Time

To assure that the liquid and gas reach equilibrium at separator pressure a certain liquid storage is required. This is defined as "retention time" or the average time a molecule of liquid is retained in the vessel assuming plug flow. The retention time is thus the volume of the liquid storage in the vessel divided by the liquid flow rate.

For most applications retention times of between 30 seconds and 3 minutes have been found to be sufficient. Where foaming crude is present retention times up to four times this amount may be needed.

Re-entrainment

Re-entrainment is a phenomenon caused by high gas velocity at the gas-liquid interface of a separator. Momentum transfer from the gas to the liquid causes waves and ripples in the liquid, and then droplets are broken away from the liquid phase.

The general rule of thumb that calls for limiting the slenderness ratio to a maximum of 4 or 5 is applicable for half-full horizontal separators. Re-entrainment should be particularly considered for high-pressure separators sized on gas-capacity constraints. It is more likely at higher operating pressures (>1,000 psig) and higher oil viscosities (<30° API). For more specific limits, see Viles [1].

SEPARATOR SIZING

Horizontal Separators

For sizing a horizontal separator it is necessary to choose a seam-to-seam vessel length and a diameter. This choice must satisfy the conditions for gas capacity that allow the liquid drops to fall from the gas to the liquid volume as the gas traverses the effective length of the vessel. It must also provide sufficient retention time to allow the liquid to reach equilibrium.

For a vessel 50% full of liquid, and separation of 100-micron liquid drops from the gas, the following equations apply:

Gas Capacity

$$d\,L_{eff} = 420\left[\frac{TZQ_g}{P}\right]\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2} \qquad (4-6)$$

where d = vessel internal diameter, in.
L_{eff} = effective length of the vessel where separation occurs, ft
T = operating temperature, °R
Q_g = gas flow rate, MMscfd
P = operating pressure, psia
Z = gas compressibility
C_D = drag coefficient
d_m = liquid drop to be separated, micron
ρ_g = density of gas, lb/ft^3
ρ_l = density of liquid, lb/ft^3

Derivation of Equation 4-6

Assume vessel is one-half full of liquid. Determine gas velocity, V_g. A is in ft^2, D in ft, d in inches, Q in ft^3/s

$$V_g = \frac{Q}{A_g}$$

$$A_g = \frac{1}{2}\left(\frac{\pi}{4}D^2\right) = \frac{1}{2}\left(\frac{\pi}{4}\frac{d^2}{144}\right) = \frac{d^2}{367}$$

Q_g is in MMscfd

$$Q = Q_g \times 10^6 \frac{\text{scf}}{\text{MMscf}} \times \frac{\text{day}}{24\text{ hr}} \times \frac{\text{hr}}{3{,}600\text{s}} \times \frac{14.7}{P} \times \frac{TZ}{520}$$

$$= 0.327\frac{TZ}{P}Q_g$$

$$V_g = \frac{\left(0.327\frac{TZ}{P}Q_g\right)(367)}{d^2}$$

$$V_g = 120 \frac{TZQ_g}{Pd^2}$$

Set the residence time of the gas equal to the time required for the droplet to fall to the gas-liquid interface:

$$t_g = \frac{L_{eff}}{V_g} \quad t_d = \frac{D}{2V_t} = \frac{d}{24V_t}$$

$$t_g = \frac{L_{eff}}{120\left(\frac{TZQ_g}{Pd^2}\right)}$$

$$\text{Recalling that } V_t = 0.0119\left[\left(\frac{\rho_l - \rho_g}{\rho_g}\right)\frac{d_m}{C_D}\right]^{1/2}$$

$$t_d = \frac{d}{(24)(0.0119)}\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}$$

Setting $t_g = t_d$,

$$\frac{L_{eff}}{120\left(\frac{TZQ_g}{Pd^2}\right)} = \frac{d\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}}{(24)(0.0119)}$$

$$L_{eff}\, d = 420 \frac{TZQ_g}{P}\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}$$

Liquid Capacity

$$d^2 L_{eff} = \frac{t_r Q_l}{0.7} \qquad (4-7)$$

where t_r = desired retention time for the liquid, min
Q_l = liquid flow rate, bpd

Derivation of Equation 4-7

t is in s, Vol in ft^3, Q in ft^3/s

$$t = \frac{Vol}{Q}$$

$$Vol = \frac{1}{2}\left(\frac{\pi D^2 L_{eff}}{4}\right) = \frac{\pi d^2 L_{eff}}{(2)(4)(144)} = 2.73 \times 10^{-3}\, d^2 L_{eff}$$

Q_l is in bpd

$$Q = Q_l \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24\ hr} \times \frac{hr}{3{,}600s} = 6.49 \times 10^{-5} Q_l$$

$$t = 42.0 \frac{d^2 L_{eff}}{Q_l}$$

$$d^2 L_{eff} = \frac{t_r Q_l}{0.7}$$

Seam-to-Seam Length and Slenderness Ratio

The seam-to-seam length of the vessel should be determined from the geometry once an effective length has been determined. Allowance must be made for the inlet diverter and mist extractor. For screening purposes the following approximation has been proven useful:

$$L_{ss} = L_{eff} + \frac{d}{12} \quad \text{for gas capacity}$$

$$L_{ss} = \frac{4}{3} L_{eff} \quad \text{for liquid capacity} \qquad (4-8)$$

Equations 4-6 and 4-7 allow for various choices of diameter and length. It can be shown that the smaller the diameter the less the vessel will weigh and thus the lower its cost. There is a point, however, where decreasing the diameter increases the possibility that high velocity in the gas flow will create waves and re-entrain liquids at the gas-liquid interface. Experience has shown that if the gas capacity governs and the length divided by the diameter (slenderness ratio) is greater than 4 or 5, re-entrainment could become a problem. Equation 4-8 indicates that slenderness ratios must be at least 1 or more. Most common separators are designed for slenderness ratios of 3 to 4.

Procedure for Sizing Horizontal Separators

1. Calculate values of d, L_{eff} that satisfy the gas capacity constraint.

$$L_{eff}d = 420\,\frac{TZQ_g}{P}\left[\frac{\rho_g}{\rho_l - \rho_g}\,\frac{C_D}{d_m}\right]^{1/2}$$

2. Calculate values of d, L_{eff} that satisfy the retention time constraint.

$$d^2L_{eff} = \frac{t_r\,Q_l}{0.7}$$

3. Estimate seam-to-seam length.

$$L_{ss} = L_{eff} + \frac{d}{12} \quad \text{for gas capacity}$$

$$L_{ss} = \frac{4}{3}\,L_{eff} \quad \text{for liquid capacity}$$

4. Select a size of reasonable diameter and length. Slenderness ratios (12 L_{ss}/d) on the order of 3 to 4 are common. Do not exceed a slenderness ratio of 5 without further study of re-entrainment.

For separators other than 50% full of liquid, equations can be derived similarly, using the actual gas and liquid areas to calculate gas velocity and liquid volume. The equations are derived using the same principles.

Vertical Separators

In vertical separators, a minimum diameter must be maintained to allow liquid drops to separate from the vertically moving gas. The liquid retention time requirement specifies a combination of diameter and liquid volume height. Any diameter greater than the minimum required for gas capacity can be chosen. Figure 4-15 shows the model used for a vertical separator.

Gas Capacity

$$d^2 = 5{,}040\left[\frac{TZQ_g}{P}\right]\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2} \qquad (4-9)$$

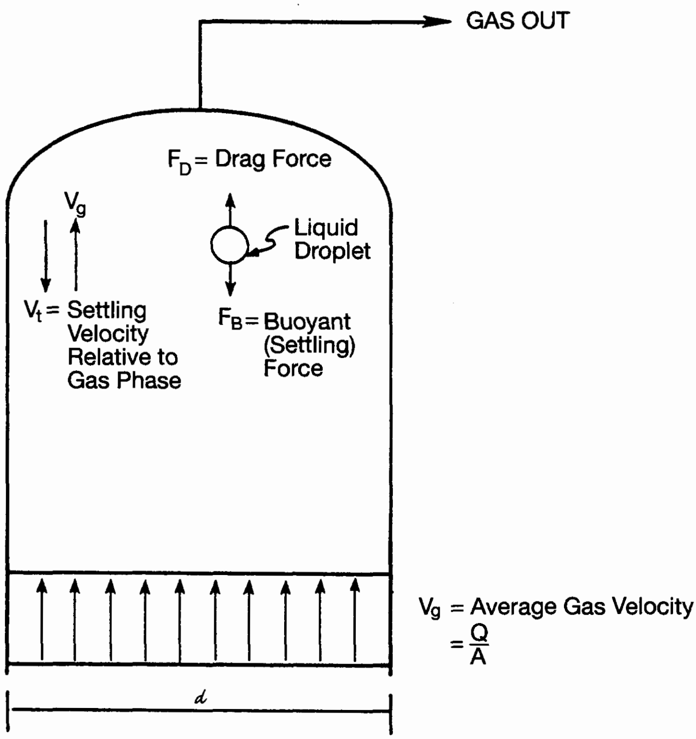

Figure 4-15. Model of a vertical separator.

Derivation of Equation 4-9

For the droplets to fall, the gas velocity must be less than the terminal velocity of the droplet. Recall that:

$$V_t = 0.0119\left[\left(\frac{\rho_l - \rho_g}{\rho_g}\right)\frac{d_m}{C_D}\right]^{1/2}$$

Determine gas velocity, V_g. A is in ft^2, D in ft, d in inches, Q in ft^3/s.

$$V_g = \frac{Q}{A_g}$$

$$A_g = \left(\frac{\pi}{4}D^2\right) = \left(\frac{\pi}{4}\frac{d^2}{144}\right) = \frac{d^2}{183}$$

Q_g is in MMscfd

$$Q = Q_g \times 10^6 \frac{\text{scf}}{\text{MMscf}} \times \frac{\text{day}}{24\text{ hr}} \times \frac{\text{hr}}{3{,}600\text{s}} \times \frac{14.7}{P} \times \frac{TZ}{520}$$

$$= 0.327 \frac{TZ}{P} Q_g$$

$$V_g = \frac{\left(0.327 \frac{TZ}{P} Q_g\right)(183)}{d^2}$$

$$V_g = 60 \frac{TZQ_g}{Pd^2}$$

$$V_t = V_g$$

$$0.0119 \left[\left(\frac{\rho_l - \rho_g}{\rho_g}\right) \frac{d_m}{C_D}\right]^{1/2} = \frac{60TZQ_g}{Pd^2}$$

$$d^2 = 5{,}040 \frac{TZQ_g}{P} \left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right) \frac{C_D}{d_m}\right]^{1/2}$$

Liquid Capacity

$$d^2\, h = \frac{t_r\, Q_l}{0.12} \qquad (4-10)$$

where h = height of the liquid volume, in.

Derivation of Equation 4-10

t is in s, Vol in ft^3, Q in ft^3/s, h in inches

$$t = \frac{\text{Vol}}{Q}$$

$$\text{Vol} = \frac{\pi\, D^2\, h}{(4)(12)} = \frac{\pi d^2\, h}{(4)(144)(12)} = 4.55 \times 10^{-4}\, d^2 h$$

Q_l is in bpd

$$Q = Q_l \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24\,hr} \times \frac{hr}{3,600s} = 6.49 \times 10^{-5} Q_l$$

$$t = 7.00 \frac{d^2h}{Q_l}$$

t_r is in min

$$d^2h = \frac{t_r\,Q_l}{0.12}$$

Seam-to-Seam Length and Slenderness Ratio

The seam-to-seam length of the vessel should be determined from the geometry once a diameter and height of liquid volume are known.

As shown in Figure 4-16, allowance must be made for the gas separation section and mist extractor and for any space below the water outlet. For screening purposes the following approximation has been proven useful. Use the larger of the two values:

$$L_{ss} = \frac{h + 76}{12} \text{ or } L_{ss} = \frac{h + d + 40}{12} \qquad (4-11)$$

As with horizontal separators, the larger the slenderness ratio, the less expensive the vessel. In vertical separators whose sizing is liquid dominated, it is common to chose slenderness ratios no greater than 4 to keep the height of the liquid collection section to a reasonable level. Choices of between 3 and 4 are common, although height restrictions may force the choice of a lower slenderness ratio.

EXAMPLES

Example 4-1: Sizing a Vertical Separator

Given: Flow Rate: 10 MMscfd at 0.6 specific gravity
2,000 bopd at 40° API
Operating Pressure: 1,000 psia
Operating Temperature: 60°F

Solution:

1. Calculate C_D

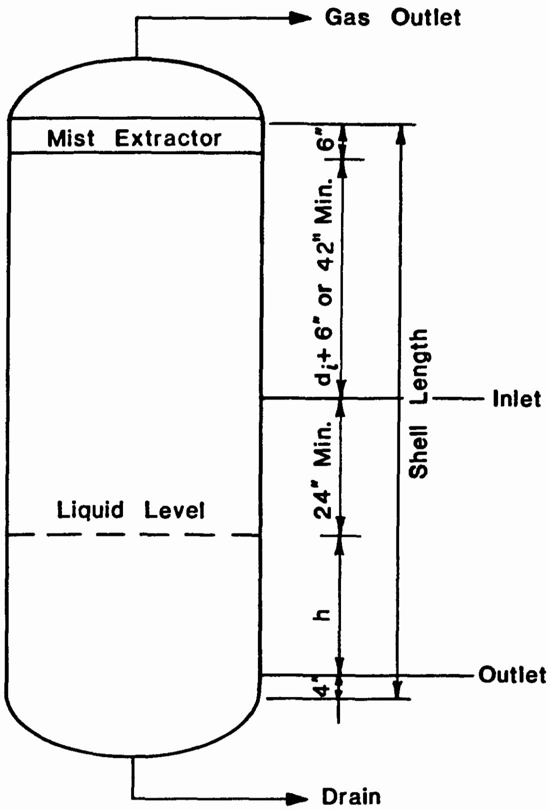

Figure 4-16. Approximate shell length from liquid level height.

$$\rho_l = 62.4[141.5/(131.5 + 40)] = 51.5 \text{ lb/ft}^3$$

$$\rho_g = 2.70 \frac{S\,P}{T\,Z}$$

$$Z = 0.84 \text{ (from Chapter 3)}$$

$$\rho_g = 2.70 \frac{(0.6)(1{,}000)}{(520)(0.84)} = 3.71$$

$$d_m = 140 \text{ micron}$$

$\mu = 0.013$ cp (from Chapter 3)

Assume $C_D = 0.34$

$$V_t = 0.0119\left[\left(\frac{51.5-3.71}{3.71}\right)\frac{140}{0.34}\right]^{1/2}$$

$V_t = 0.866$

$$Re = 0.0049\left[\frac{(3.71)(140)(0.866)}{0.013}\right] = 169.54$$

$$C_D = \frac{24}{169.54} + \frac{3}{(169.54)^{1/2}} + 0.34$$

$C_D = 0.711$

Repeat using $C_D = 0.711$

$V_t = 0.597$

$Re = 118$

$C_D = 0.820$

Repeat:

$V_t = 0.556$

$Re = 110$

$C_D = 0.844$

Repeat:

$V_t = 0.548$

$Re = 108$

$C_D = 0.851$

Repeat:

$V_t = 0.545$

$Re = 108$

$C_D = 0.851$ OK

2. Gas capacity constraint

$$d^2 = 5{,}040\left[\frac{TZQ_g}{P}\right]\left[\left(\frac{\rho_g}{\rho_l-\rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}$$

$Z = 0.84$ (from Chapter 3)

$$d^2 = 5,040\left[\frac{(520)(0.84)(10)}{1,000}\right]\left[\left(\frac{3.71}{51.5-3.71}\right)\frac{0.851}{140}\right]^{1/2}$$

$d = 21.8$ in.

3. Liquid capacity constraint

$$d^2 h = \frac{t_r Q_l}{0.12}$$

4. Compute combinations of d and h for various t_r (Table 4-1).
5. Compute seam-to-seam length (Table 4-1).

$$L_{ss} = \frac{h+76}{12} \text{ or } L_{ss} = \frac{h+d+40}{12}$$

where d is the minimum diameter for gas capacity

6. Compute slenderness ratio ($12\ L_{ss}/d$). Choices in the range of 3 to 4 are most common (Table 4-1).
7. Choose a reasonable size with a diameter greater than that determined by the gas capacity. A 36-in. × 10-ft separator provides slightly more than three minutes retention time with a diameter greater than 21.8 in. and a slenderness ratio of 3.2.

Table 4-1
Vertical Separator Example
Diameter vs. Length for Liquid Capacity Constraint

t_r min	d in.	h in.	L_{ss} ft	$(12)L_{ss}/d$
3	24	86.8	13.6	6.8
	30	55.6	11.0	4.4
	36	38.6	9.6	3.2
	42	28.3	8.7	2.5
	48	21.7	8.1	2.0
2	24	57.9	11.2	5.6
	30	37.0	9.4	3.8
	36	25.7	8.5	2.8
	42	18.9	7.9	2.3
1	24	28.9	8.7	4.4
	30	18.5	7.9	3.2
	36	12.9	7.4	2.5

Example 4-2: Sizing a Horizontal Separator

Given: Flow Rate: 10 MMscfd at 0.6 specific gravity
2,000 bopd at 40° API
Operating Pressure: 1,000 psia
Operating Temperature: 60° F

Solution:

1. Calculate C_D (same as Example 4-1)

   $C_D = 0.851$

2. Gas capacity constraint

$$d\,L_{eff} = 420\left[\frac{TZQ_g}{P}\right]\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}$$

   Z = 0.84 (from Chapter 3)

$$d\,L_{eff} = 420\left[\frac{(520)\,(0.84)\,(10)}{1{,}000}\right]\left[\left(\frac{3.71}{51.5 - 3.71}\right)\frac{0.851}{140}\right]^{1/2}$$

   $d\,L_{eff} = 55.04$

3. Liquid capacity constraint

$$d^2L_{eff} = \frac{t_r\,Q_l}{0.7}$$

4. Compute combinations of d and L_{ss} for gas and liquid capacity.
5. Compute seam-to-seam length for various d (Table 4-2).

$$L_{ss} = L_{eff} + \frac{d}{12}$$

6. Compute slenderness ratios (12 L_{ss}/d). Choices in the range of 3 to 4 are common.
7. Choose a reasonable size with a diameter and length combination above both the gas capacity and the liquid capacity constraint lines. A 36-in. × 10-ft separator provides about 3 minutes retention time.

Table 4-2
Horizontal Separator Example
Diameter vs. Length

d	Gas L_{eff}	Liquid L_{eff}	L_{ss}	$12L_{ss}/d$
16	2.5	33.5	44.7	33.5
20	2.0	21.4	28.5	17.1
24	1.7	14.9	19.9	9.9
30	1.3	9.5	12.7	5.1
36	1.1	6.6	9.1*	3.0
42	0.9	4.9	7.4*	2.1
48	0.8	3.7	6.2*	1.6

*$L_{ss} = L_{eff} + 2.5$ governs

REFERENCES

1. Viles J. C. "Predicting Liquid Re-entrainment in Horizontal Separators" (SPE 25474). Paper presented at the Production Operations Symposium in Oklahoma City, OK, USA, in March 1993.

CHAPTER

5

Oil and Water Separation*

INTRODUCTION

This chapter discusses the concepts, theory, and sizing equations for the separation of two immiscible liquid phases (in this case, normally crude oil and produced water). The separator design concepts that have been presented in Chapter 4 relate to the two-phase separation of liquid and gas and are applicable to the separation of gas that takes place in three-phase separators, gas scrubbers, and any other device in which gas is separated from a liquid phase.

When oil and water are mixed with some intensity and then allowed to settle, a layer of relatively clean free water will appear at the bottom. The growth of this water layer with time will follow a curve as shown in Figure 5-1. After a period of time, ranging anywhere from three minutes to thirty minutes, the change in the water height will be negligible. The water fraction, obtained from gravity settling, is called "free water." It is normally beneficial to separate the free water before attempting to treat the remaining oil and emulsion layers.

"Three-phase separator" and "free-water knockout" are terms that are used to describe pressure vessels that are designed to separate and remove

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*Reviewed for the 1998 edition by Mary E. Thro of Paragon Engineering Services, Inc.

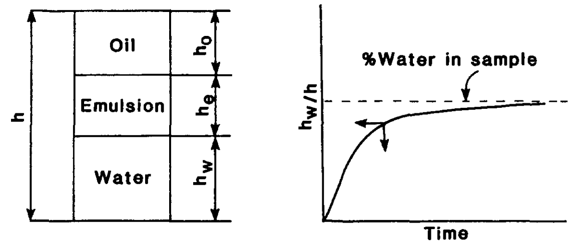

Figure 5-1. Growth of water layer with time.

the free water from a mixture of crude oil and water. Because flow normally enters these vessels either directly from (1) a producing well or (2) a separator operating at a higher pressure, the vessel must be designed to separate the gas that flashes from the liquid as well as separate the oil and water.

The term "three-phase separator" is normally used when there is a large amount of gas to be separated from the liquid, and the dimensions of the vessel are determined by the gas capacity equations discussed in Chapter 4. "Free-water knockout" is generally used when the amount of gas is small relative to the amount of oil and water, and the dimensions of the vessel are determined by the oil/water separation equations discussed in this chapter. No matter what name is given to the vessel, any vessel that is designed to separate two immiscible liquid phases will employ the concepts described in this chapter. For purposes of this chapter, we will call such a vessel a "three-phase separator."

The basic design aspects of three-phase separation are identical to those discussed in Chapter 4. The only additions are that more concern is placed on liquid-liquid settling rates; and that some means of removing the free water must be added. Liquid-liquid settling rates will be discussed later in this chapter. Water removal is a function of the control methods used to maintain separation and removal from the oil. Several control methods are applicable to three-phase separators. The shape and diameter of the vessel will, to a degree, determine the types of control used.

EQUIPMENT DESCRIPTION

Horizontal Separators

Three-phase separators are designed as either horizontal or vertical pressure vessels. Figure 5-2 is a schematic of a horizontal separator. The

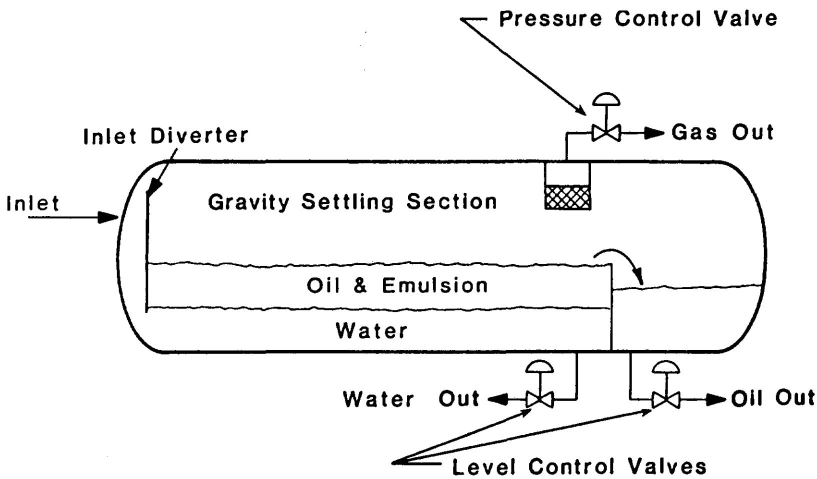

Figure 5-2. Horizontal three-phase separator schematic.

fluid enters the separator and hits an inlet diverter. This sudden change in momentum does the initial gross separation of liquid and vapor as discussed in Chapter 4. In most designs, the inlet diverter contains a downcomer that directs the liquid flow below the oil/water interface.

This forces the inlet mixture of oil and water to mix with the water continuous phase in the bottom of the vessel and rise through the oil/water interface. This process is called "water-washing," and it promotes the coalescence of water droplets which are entrained in the oil continuous phase. The inlet diverter assures that little gas is carried with the liquid, and the water wash assures that the liquid does not fall on top of the gas/oil or oil/water interface, mixing the liquid retained in the vessel and making control of the oil/water interface difficult.

The liquid collecting section of the vessel provides sufficient time so that the oil and emulsion form a layer or "oil pad" at the top. The free water settles to the bottom. Figure 5-2 illustrates a typical horizontal separator with an interface controller and weir. The weir maintains the oil level and the level controller maintains the water level. The oil is skimmed over the weir. The level of the oil downstream of the weir is controlled by a level controller that operates the oil dump valve.

The produced water flows from a nozzle in the vessel located upstream of the oil weir. An interface level controller senses the height of the oil/water interface. The controller sends a signal to the water dump

valve thus allowing the correct amount of water to leave the vessel so that the oil/water interface is maintained at the design height.

The gas flows horizontally and out through a mist extractor to a pressure control valve that maintains constant vessel pressure. The level of the gas/oil interface can vary from half the diameter to 75% of the diameter depending on the relative importance of liquid/gas separation. The most common configuration is half full, and this is used for the design equations in this section. Similar equations can be developed for other interface levels.

Figure 5-3 shows an alternate configuration known as a "bucket and weir" design. This design eliminates the need for a liquid interface controller. Both the oil and water flow over weirs where level control is accomplished by a simple displacer float. The oil overflows the oil weir into an oil bucket where its level is controlled by a level controller that operates the oil dump valve. The water flows under the oil bucket and then over a water weir. The level downstream of this weir is controlled by a level controller that operates the water dump valve.

The height of the oil weir controls the liquid level in the vessel. The difference in height of the oil and water weirs controls the thickness of the oil pad due to specific gravity differences. It is critical to the operation of the vessel that the water weir height be sufficiently below the oil weir height so that the oil pad thickness provides sufficient oil retention time. If

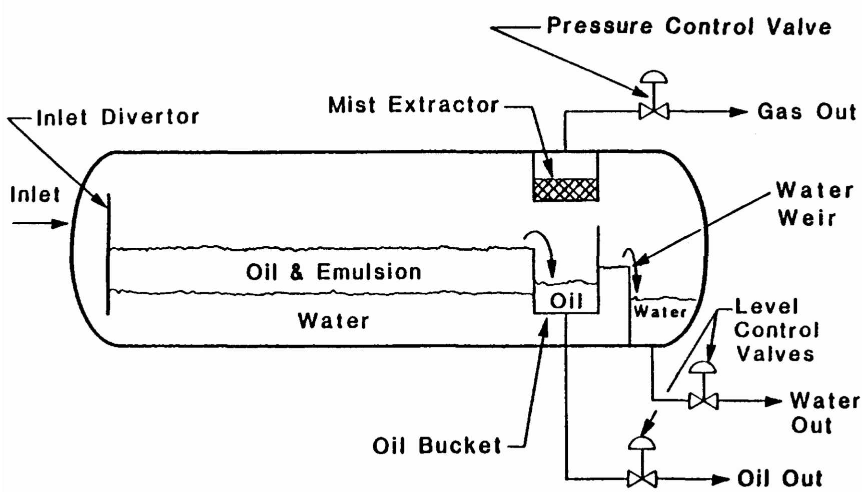

Figure 5-3. Bucket and weir design.

the water weir is too low and the difference in specific gravity is not as great as anticipated, then the oil pad could grow in thickness to a point where oil will be swept under the oil box and out the water outlet. Normally, either the oil or the water weir is made adjustable so that changes in oil/water specific gravities or flow rates can be accommodated.

To obtain a desired oil pad height, the water weir should be set a distance below the oil weir, which is calculated by:

$$\Delta h = h_o \left[1 - \left(\frac{\rho_o}{\rho_w}\right)\right] \tag{5-1}$$

where Δh = distance below the oil weir, in.
h_o = desired oil pad height, in.
ρ_o = oil density, lb/ft^3
ρ_w = water density, lb/ft^3

This equation neglects the height of the oil and water flowing over the weir and presents a view of the levels when there is no inflow. A large inflow of oil will cause the top of the oil pad to rise; the oil pad will thus get thicker, and the oil bucket must be deep enough so that oil does not flow under it. Similarly, a large inflow of water will cause the level of water flowing over the water weir to rise, and there will be a large flow of oil from the oil pad over the oil weir until a new h_w is established. These dynamic effects can be minimized by making the weirs as long as possible.

Derivation of Equation 5-1

ρ is in lb/ft^3, h is in inches. Setting the pressures at Point "A" in Figure 5-4 equal,

$$\rho_o h_o + \rho_w h_w = \rho_w h_w'$$

$$h_w = \frac{\rho_w h_w' - \rho_o h_o}{\rho_w} = h_w' - \frac{\rho_o}{\rho_w} h_o$$

$$\Delta h = h_o + h_w - h_w'$$

$$\Delta h = h_o - \frac{\rho_o}{\rho_w} h_o = h_o\left[1 - \frac{\rho_o}{\rho_w}\right]$$

Interface control has the advantage of being easily adjustable to handle unexpected changes in oil or water specific gravity or flow rates. However, in heavy oil applications or where large amounts of emulsion or paraf-

fin are anticipated it may be difficult to sense interface level. In such a case bucket and weir control is recommended.

In some areas of the world, the term "free-water knockout" is reserved for a vessel which processes an inlet liquid stream with little entrained gas and makes no attempt to separate the gas from the oil. Such a vessel has only an oil outlet and a water outlet (no separate gas outlet), as shown in Figure 5-5. It should be clear that the principles of operation of such a vessel are the same as those described above.

Vertical Separators

Figure 5-6 shows a typical configuration for a vertical three-phase separator. Flow enters the vessel through the side as in the horizontal separator, the inlet diverter separates the bulk of the gas. A downcomer is

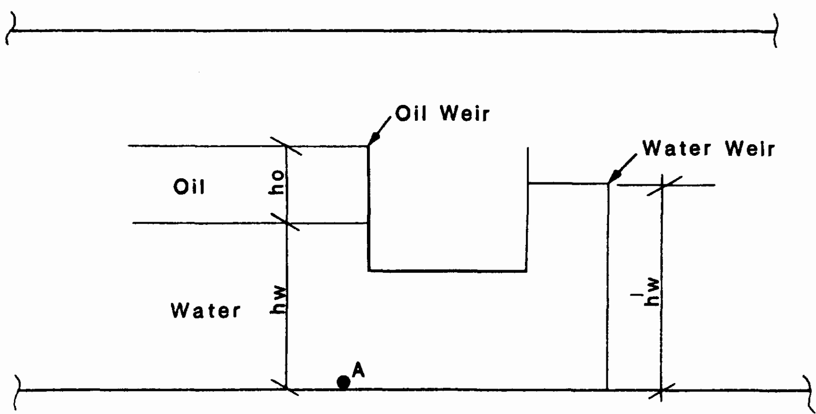

Figure 5-4. Determination of oil pad height.

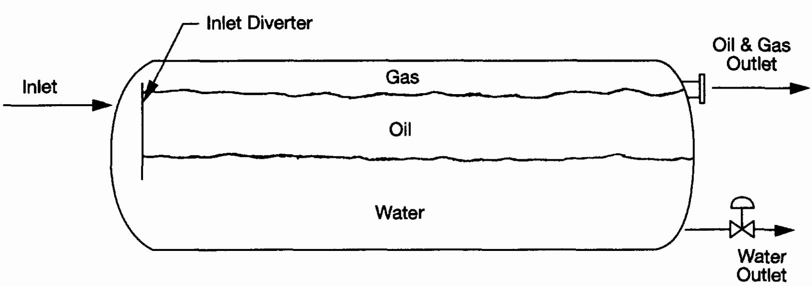

Figure 5-5. Free-water knockout (FWKO).

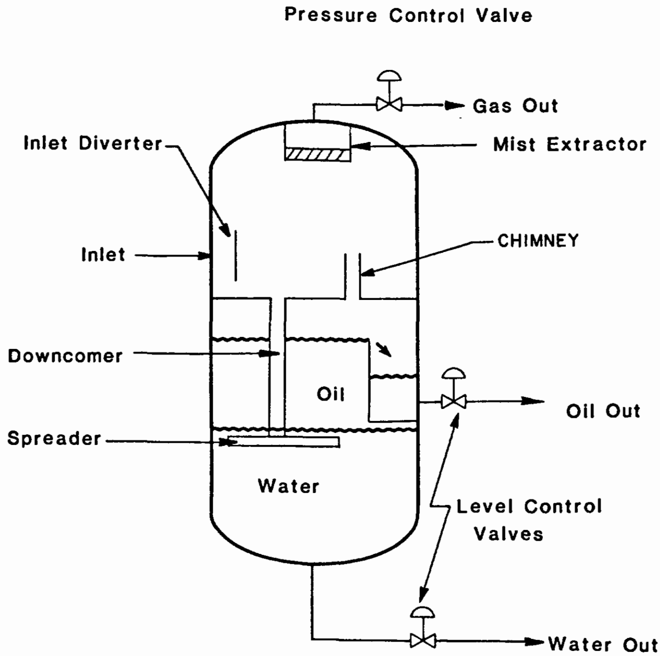

Figure 5-6. Vertical three-phase separator schematic.

required to transmit the liquid through the oil-gas interface so as not to disturb the oil skimming action taking place. A chimney is needed to equalize gas pressure between the lower section and the gas section.

The spreader or downcomer outlet is located at the oil-water interface. From this point as the oil rises any free water trapped within the oil phase separates out. The water droplets flow countercurrent to the oil. Similarly, the water flows downward and oil droplets trapped in the water phase tend to rise countercurrent to the water flow.

Sometimes a cone bottom three-phase separator is used. This is a design that would be used if sand production was anticipated to be a major problem. The cone is normally at an angle to the horizontal of between 45° and 60°. Produced sand may have a tendency to stick to steel at 45°. If a cone is installed it could be part of the pressure containing walls of the vessel, or for structural reasons, it could be installed internal to the vessel cylinder. In such a case, a gas equalizing line must be installed to assure that the vapor behind the cone is always in pressure equilibrium with the vapor space.

Figure 5-7 shows the three different methods of control that are often used on vertical separators.

The first is strictly level control. A regular displacer float is used to control the gas-oil interface and regulate a control valve dumping oil from the oil section. An interface float is used to control the oil-water interface and regulate a water outlet control valve. Because no internal baffling or weirs are used, this system is the easiest to fabricate and handles sand and solids production best.

The second method shown uses a weir to control the gas-oil interface level at a constant position. This results in a better separation of water from the oil as all the oil must rise to the height of the oil weir before exiting the vessel. Its disadvantages are that the oil box takes up vessel volume and costs money to fabricate. In addition, sediment and solids could collect in the oil box and be difficult to drain, and a separate low level shut-down may be required to guard against the oil dump valve failing to open.

The third method uses two weirs, which eliminates the need for an interface float. Interface level is controlled by the height of the external water weir relative to the oil weir or outlet height. This is similar to the bucket and weir design of horizontal separators. The advantage of this system is that it eliminates the interface level control. The disadvantage is that it requires additional external piping and space.

Horizontal vs. Vertical Selection

The benefits of each type of design were described earlier. As in two-phase separation, it is also true for three-phase separation that the flow

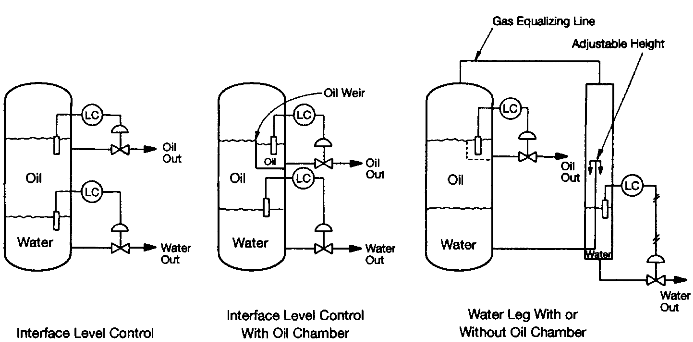

Figure 5-7. Liquid level control schemes.

geometry in a horizontal vessel is more favorable from a process standpoint. However, there may be non-process reasons to select a vertical vessel for a specific application.

VESSEL INTERNALS

Most of the vessel internals are discussed in Chapter 4. Two common internals not discussed are coalescing plates and sand jets. It is possible to use various plate or pipe coalescer designs to aid in the coalescing of oil droplets in the water and water droplets in the oil.

Coalescing Plates

The installation of coalescing plates in the liquid section will cause the size of the water droplets entrained in the oil phase to increase, making gravity settling of these drops to the oil-water interface easier. Thus, the use of coalescing plates or the use of free-flow turbulent coalescers (SP Packs), which are both described in Chapter 7, will often lead to the ability to handle a given flow rate in a smaller vessel. However, because of the potential for plugging with sand, paraffin, or corrosion products, the use of coalescing plates should be discouraged, except for instances where the savings in vessel size and weight are large enough to justify the potential increase in operating costs and decrease in availability.

Sand Jets and Drains

In horizontal three-phase separators, one worry is the accumulation of sand and solids at the bottom of the vessel. If allowed to build up, these solids upset the separator operations by taking up vessel volume. Generally, the solids settle to the bottom and become well packed.

To remove the solids, sand drains are opened in a controlled manner, and then high-pressure fluid, usually produced water, is pumped through the jets to agitate the solids and flush them down the drains. The sand jets are normally designed with a 20 ft/s jet tip velocity and aimed in such a manner to give good coverage of the vessel bottom.

To prevent the settled sand from clogging the sand drains, sand pans or sand troughs are used to cover the outlets. These are inverted troughs with slotted side openings.

EMULSIONS

Emulsions can be particularly troublesome in the operation of three-phase separators. Over a period of time an accumulation of emulsified materials and/or other impurities usually will form at the interface of the water and oil phases. In addition to adverse effects on the liquid level control, this accumulation will also decrease the effective oil or water retention time in the separator, with a resultant decrease in water-oil separation efficiency. Addition of chemicals and/or heat often minimizes this difficulty.

Frequently, it is possible to appreciably lower the settling time necessary for water-oil separation by either the application of heat in the liquid section of the separator or the addition of de-emulsifying chemicals. The treating of emulsions is discussed in more detail in Chapter 6.

THEORY

Gas Separation

The concepts and equations pertaining to two-phase separation described in Chapter 4 are equally valid for three-phase separation.

Oil/Water Settling

It can be shown that flow around settling oil drops in water or water drops in oil is laminar and thus Stokes' Law governs. The terminal drop velocity is:

$$V_t = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)\,d_m^2}{\mu} \tag{5-2}$$

where V_t = terminal settling velocity, ft/s
$\Delta S.G.$ = difference in specific gravity relative to water between the oil and the water phases
d_m = drop size, micron
μ = viscosity of continuous phase, cp

Water Droplet Size in Oil

It is difficult to predict the water droplet size that must be settled out of the oil phase to coincide with the rather loose definition of "free oil." Unless laboratory or nearby field data are available, good results have

been obtained by sizing the oil pad such that water droplets 500 microns and larger settle out. If this criteria is met, the emulsion to be treated by downstream equipment should contain less than 5% to 10% water without an excessive chemical treatment program.

Oil Droplet Size in Water

From Equation 5-2 it can be seen that the separation of oil droplets from the water is easier than the separation of water droplets from the oil. The oil's viscosity is on the order of 50 to 20 times that of water. The primary purpose of three-phase separation is to prepare the oil for further treating. Field experience indicates that oil content in the produced water from a three-phase separator, sized for water removal from oil, can be expected to be between a few hundred and 2,000 mg/l. This water will require further treating and is discussed later. Sizing for oil droplet removal from the water phase does not appear to be a meaningful criterion.

Retention Time

A certain amount of oil storage is required to assure that the oil reaches equilibrium and flashed gas is liberated. An additional amount of storage is required to assure that the free water has time to coalesce into droplet sizes sufficient to fall in accordance with Equation 5-2. It is common to use retention times ranging from three minutes to thirty minutes depending upon laboratory or field data. If this information is not available, an oil retention time of ten minutes is suggested for design.

Similarly, a certain amount of water storage is required to assure that most of the large droplets of oil entrained in the water have sufficient time to coalesce and rise to the oil-water interface. It is common to use retention times for the water phase ranging from three minutes to thirty minutes depending upon laboratory or field data. If this information is not available, a water retention time of ten minutes is recommended for design.

The retention time for both the maximum oil rate and the maximum water rate should be calculated, unless laboratory data indicate that it is unnecessary to take this conservative design approach.

SEPARATOR SIZING

The guidelines presented here can be used for initial sizing determinations. They are meant to complement and not replace operating experi-

ences. Determination of the type and size separator must be made on an individual basis. All the functions and requirements should be considered including the likely uncertainties in design flow rates and properties. For this reason, there is no substitute for good engineering evaluations of each separator by the design engineer. The "trade off" between design size and details and uncertainties in design parameters should not be left to manufacturer recommendations or rules of thumb.

Horizontal Separators

For sizing a horizontal three-phase separator it is necessary to specify a vessel diameter and a seam-to-seam vessel length. The gas capacity and retention time considerations establish certain acceptable combinations of diameter and length. The need to settle 500-micron water droplets from the oil establishes a maximum diameter.

Gas Capacity

The gas capacity constraints provide the following formula, discussed in Chapter 4:

$$d^2 L_{eff} = 420 \left[\frac{TZQ_g}{P}\right] \left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right) \frac{C_D}{d_m}\right]^{1/2} \qquad (5-3)$$

where d = vessel inside diameter, in.
L_{eff} = vessel effective length, ft
T = operating temperature, °R
Z = gas compressibility
Q_g = gas flow rate, MMscfd
ρ = operating pressure, psia
ρ_g = density of gas, lb/ft^3
ρ_l = density of liquid, lb/ft^3
C_D = drag coefficient
d_m = liquid drop to be separated, microns

Retention Time

Retention time constraints give another equation that provides acceptable combinations of d and L_{eff}.

$$d^2 L_{eff} = 1.42 \left[(Q_w)(t_r)_w + (Q_o)(t_r)_o\right] \qquad (5-4)$$

where Q_w = water flow rate, bpd
$(t_r)_w$ = water retention time, min
Q_o = oil flow rate, bpd
$(t_r)_o$ = oil retention time, min

Derivation of Equation 5-4

t is in S, Vol in ft^3, Q in ft^3/s, D in ft, d in inches, L_{eff} in ft

$$t = \frac{Vol}{Q}$$

$$Vol = \frac{1}{2}\left(\frac{\pi D^2 L_{eff}}{4}\right) = \frac{\pi d^2 L_{eff}}{(2)(4)(144)} = 2.73 \times 10^{-3} d^2 L_{eff}$$

$$(Vol)_o = 2.73 \times 10^{-3} d^2 L_{eff}\left(\frac{A_o}{A_l}\right)$$

$$(Vol)_w = 2.73 \times 10^{-3} d^2 L_{eff}\left(\frac{A_w}{A_l}\right)$$

Q_o and Q_w are in bpd

$$Q = Q_o \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24\ hr} \times \frac{hr}{3{,}600s} = 6.49 \times 10^{-5} Q_o$$

$$Q = 6.49 \times 10^{-5}\ Q_w$$

A_o, A_w, and A_l are cross-sectional areas of oil, water, and liquid.

$$42\left(\frac{A_o}{A_l}\right) = \frac{t_o Q_o}{d^2 L_{eff}} \quad 42\left(\frac{A_w}{A_l}\right) = \frac{t_w Q_w}{d^2 L_{eff}}$$

$(t_r)_o$ and $(t_r)_w$ are in minutes

$$0.7\left(\frac{A_o}{A_l}\right) = \frac{(t_r)_o Q_o}{d^2 L_{eff}} \quad 0.7\left(\frac{A_w}{A_l}\right) = \frac{(t_r)_w Q_w}{d^2 L_{eff}}$$

$$0.7\left(\frac{A_o + A_w}{A_l}\right) = \frac{(t_r)_o Q_o + (t_r)_w Q_w}{d^2 L_{eff}}$$

$$d^2 L_{eff} = 1.42\ [(t_r)_o Q_o + (t_r)_w Q_w]$$

Settling Equation

The requirement that 500-micron water droplets be capable of settling out of the oil pad establishes a maximum oil pad thickness given by the following formula:

$$h_o = \frac{0.00128\,(t_r)_o\,(\Delta S.G.)\,d_m^2}{\mu} \qquad (5-5)$$

Derivation of Equation 5-5

t_w, t_o are in s, V in ft/s, h_o in inches, d_m in micron, μ in cp

$t_w = t_o$

$$t_w = \frac{h_o/12}{V_t},\; V_t = \frac{1.78\times10^{-6}\,(\Delta S.G.)\,d_m^2}{\mu}$$

$$t_w = 46{,}800\,\frac{\mu\,h_o}{(\Delta S.G.)\,d_m^2}$$

t_r is in minutes

$t_o = 60\,(t_r)_o$

$$46{,}800\,\frac{\mu\,h_o}{(\Delta S.G.)\,d_m^2} = 60\,(t_r)_o$$

$$h_o = \frac{0.00128\,(t_r)_o\,(\Delta S.G.)\,d_m^2}{\mu}$$

This is the maximum thickness the oil pad can be and still allow the water droplets to settle out in time $(t_r)_o$.

For $d_m = 500$ micron

$$(h_o)_{max} = 320\,\frac{(t_r)_o\,(\Delta S.G.)}{\mu} \qquad (5-6)$$

For a given oil retention time and a given water retention time the maximum oil pad thickness constraint establishes a maximum diameter in accordance with the following procedure:

1. Compute $(h_o)_{max}$.
2. Calculate the fraction of the vessel cross-sectional area occupied by the water phase. This is given by:

$$\frac{A_w}{A} = 0.5 \frac{Q_w (t_r)_w}{(t_r)_o Q_o + (t_r)_w Q_w} \qquad (5-7)$$

Derivation of Equation 5-7

A_o and A_w are in ft^2, Q in ft^3/s, t in s, L_{eff} in ft.

$$A = \frac{Q\,t}{L_{eff}}$$

$Q = 6.49 \times 10^{-5}\, Q_o$, $Q = 6.49 \times 10^{-5}\, Q_w$

$t_o = 60(t_r)_o$, $t = 60(t_r)_w$

$$A_o = 3.89 \times 10^{-3} \frac{Q_o (t_r)_o}{L_{eff}}, \quad A_w = 3.89 \times 10^{-3} \frac{Q_w (t_r)_w}{L_{eff}}$$

$$A = 2\,(A_o + A_w)$$

$$\frac{A_w}{A} = 0.5 \frac{Q_w (t_r)_w}{(t_r)_o Q_o + (t_r)_w Q_w}$$

3. From Figure 5-8 determine the coefficient β.

4. Calculate d_{max} from:

$$d_{max} = \frac{(h_o)_{max}}{\beta}, \text{ where } \beta = h(o)/d \qquad (5-8)$$

Any combination of d and L_{eff} that satisfies all three of Equations 5-3, 5-4, and 5-5 will meet the necessary criteria.

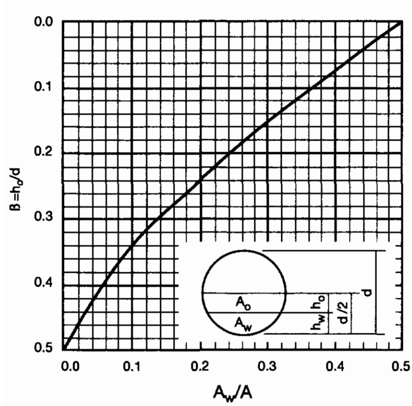

Figure 5-8. Coefficient "β" for a cylinder half filled with liquid.

Seam-to-Seam Length and Slenderness Ratios

The seam-to-seam length can be estimated from the effective length using the same formulas as for two-phase separators. Where the gas capacity governs, the slenderness ratio should be limited to less than 4 or 5 to prevent re-entrainment of liquid at the gas-liquid interface. If the separator sizing is based on liquid capacity a higher slenderness ratio is acceptable. There is the possibility of generating internal waves at the oil-water interface. Unless a more elaborate study is performed, it is recommended that slenderness ratios of less than 4 be chosen. Most common horizontal three-phase separators have slenderness ratios between 3 and 5.

Procedure for Sizing Three-Phase Horizontal Separators

1. Select a $(t_r)_o$ and a $(t_r)_w$.
2. Calculate $(h_o)_{max}$. Use 500-micron droplet if no other information is available.

$$(h_o)_{max} = 1.28 \times 10^{-3} \frac{(t_r)_o \, (\Delta S.G.) \, d_m^2}{\mu}$$

For 500 microns, $(h_o)_{max} = 320 \dfrac{(t_r)_o \, (\Delta S.G.)}{\mu}$

3. Calculate A_w/A:

$$\frac{A_w}{A} = 0.5 \frac{Q_w \, (t_r)_w}{(t_r)_o Q_o + (t_r)_w Q_w}$$

4. Determine h_o/d from curve.
5. Calculate d_{max}.

$$d_{max} = \frac{(h_o)_{max}}{h_o / d}$$

Note: d_{max} depends on Q_o, Q_w, $(t_r)_o$, and $(t_r)_w$

6. Calculate combinations of d, L_{eff} for d less than d_{max} that satisfy the gas capacity constraint. Use 100-micron droplet if no other information is available.

$$d\,L_{eff} = 420\left(\frac{TZQ_g}{P}\right)\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2}$$

7. Calculate combinations of d, L_{eff} for d less than d_{max} that satisfy the oil and water retention time constraints.

   $$d^2 L_{eff} = 1.42\,[(t_r)_o Q_o + (t_r)_w Q_w]$$

8. Estimate seam-to-seam length:

   $$L_{ss} = L_{eff} + \frac{d}{12} \text{ for gas capacity}$$

   $$L_{ss} = \frac{4}{3} L_{eff} \text{ for liquid capacity}$$

9. Select reasonable diameter and length. Slenderness ratios ($12\,L_{ss}/d$) on the order of 3 to 5 are common.

For separators other than 50% full of liquid, equations can be derived similarly, using the actual oil and water areas. The equations are derived using the same principles.

Vertical Separators

As with vertical two-phase separators, a minimum diameter must be maintained to assure adequate gas capacity. In addition, vertical three-phase separators must maintain a minimum diameter to allow the 500-micron water droplets to settle. The height of the three-phase separator is determined from retention time considerations.

Gas Capacity

The gas capacity constraints provide the following formula discussed in Chapter 4:

$$d^2_{min} = 5{,}040\left[\frac{TZQ_g}{P}\right]\left[\left(\frac{\rho_g}{\rho_l - \rho_g}\right)\frac{C_D}{d_m}\right]^{1/2} \qquad (5-9)$$

Settling

The requirement for settling water droplets from the oil requires that the following equation must be satisfied:

$$d_2 = 6{,}690 \frac{Q_o \mu}{(\Delta S.G.)\, d_m^2} \tag{5-10}$$

Derivation of Equation 5-10

V_t is in ft/s, V_o in ft/s, d_m in micron, μ in cp

$V_t = V_o$

$$V_t = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)\, d_m^2}{\mu}$$

Q is in ft³/s, A in ft²

$$V_o = \frac{Q}{A}$$

Q_o is in bpd

$$Q = Q_o \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24\ hr} \times \frac{hr}{3{,}600s} = 6.49 \times 10^{-5} Q_o$$

D is in ft, d in inches

$$A = \frac{\pi D^2}{4} = \frac{\pi d^2}{(4)(144)}$$

$$V_o = 0.0119 \frac{Q_o}{d^2}$$

$$\frac{1.78 \times 10^{-6}(\Delta S.G.)\, d_m^2}{\mu} = 0.0119 \frac{Q_o}{d^2}$$

$$d^2 = 6{,}690 \frac{Q_o \mu}{(\Delta S.G.)\, d_m^2}$$

For 500-micron droplets Equation 5-10 becomes:

$$d^2 = 0.0267 \frac{Q_o \mu}{\Delta S.G.} \tag{5-11}$$

Retention Time

$$h_o + h_w = \frac{(t_r)_o Q_o + (t_r)_w Q_w}{0.12\, d^2} \qquad (5-12)$$

where h_o = height of oil pad, in.

h_w = height from water outlet to interface, in.

(Note that this height must be adjusted for cone bottom vessels.)

Derivation of Equation 5-12

From two-phase separator design:

$$d^2\, h = \frac{t_r Q_l}{0.12}$$

Thus,

$$d^2\, h_o = \frac{(t_r)_o Q_o}{0.12}$$

$$d^2\, h_w = \frac{(t_r)_w Q_w}{0.12}$$

$$h_o + h_w = \frac{(t_r)_o Q_o + (t_r)_w Q_w}{0.12\, d^2}$$

Seam-to-Seam Length and Slenderness Ratios

As in the case of a vertical two-phase separator, the seam-to-seam length (L_{ss}) can be approximated from the geometry once h_o and h_w are chosen. For screening purposes it can be assumed that L_{ss} is the larger of the two values in the following equations:

$$L_{ss} = \frac{h_o + h_w + 76}{12} \text{ or } \frac{h_o + h_w + d + 40}{12} \qquad (5-13)$$

where d is the minimum diameter for gas capacity.

Any d larger than that calculated by Equation 5-9 and 5-10 and that satisfies Equation 5-12 is acceptable. Diameter should be chosen with slenderness ratios less than 4. Most vertical three-phase separators have slenderness ratios on the order of 1.5 to 3 to keep within reasonable height restrictions.

Procedure for Sizing Three-Phase, Vertical Separators

1. Calculate minimum diameter from requirement for water droplets to fall through oil layer. Use 500-micron droplets if no other information is available.

$$d^2 = 6{,}690 \frac{Q_o \mu}{(\Delta S.G.)\, d_m^2}$$

$$\text{For 500 - micron } d^2 = \frac{Q_o \mu}{\Delta S.G.}$$

2. Calculate minimum diameter from requirement for oil droplets to fall through gas. Use 100-micron droplets if no other information is available.

$$d^2 = 5{,}040 \frac{TZQ_g}{P} \left[\left(\frac{\rho_g}{\rho_l - \rho_g} \right) \frac{C_D}{d_m} \right]^{1/2}$$

3. Choose the larger of the two as d_{min}.
4. Select $(t_r)_o$ and $(t_r)_w$, and solve for $h_o + h_w$ for various d.

$$h_o + h_w = \frac{[(t_r)_o Q_o + (t_r)_w Q_w]}{0.12\, d^2}$$

5. Estimate seam-to-seam length using the larger value.

$$L_{ss} = \frac{h_o + h_w + 76}{12} \text{ or } \frac{h_o + h_w + d + 40}{12}$$

6. Select a size of reasonable diameter and length. Slenderness ratios ($12\, L_{ss}/d$) on the order of 1.5 to 3 are common.

EXAMPLES

Example 5-1: Sizing a Vertical Three-Phase Separator

Given: $Q_o = 5{,}000$ bopd
$Q_w = 3{,}000$ bwpd
$Q_g = 5$ MMscfd
$P_o = 100$ psia
$T_o = 90°F$

$$\text{Oil} = 30° \text{ API}$$
$$(\text{S.G.})_w = 1.07$$
$$S_g = 0.6$$
$$(t_r)_o = (t_r)_w = 10 \text{ min.}$$
$$\text{Oil viscosity} = 10 \text{ cp}$$

Solution:

1. Calculate difference in specific gravities.

$$°\text{API} = \frac{141.5}{(\text{S.G.})_o} - 131.5$$

$$(\text{S.G.})_o = \frac{141.5}{30 + 131.5}$$

$$= 0.876$$

$$\Delta\text{S.G.} = 1.07 - 0.876 = 0.194$$

2. Calculate minimum diameter to satisfy gas capacity constraint. (See two-phase separator chapter for procedure.)

$$d_{min} = 34.9 \text{ in.}$$

3. Calculate minimum diameter for water droplet settling.

$$d_{min}^2 = 6{,}690\left(\frac{Q_o\,\mu}{\Delta\text{S.G.}\; d_m^2}\right)$$

$$= 6{,}690\left[\frac{(5{,}000)(10)}{0.194\,(500)^2}\right]$$

$$d_{min} = 83.0 \text{ in.}$$

4. Liquid retention constraint.

$$h_o = \frac{(t_r)_o\,(Q_o)}{0.12\,d^2}$$

$$h_w = \frac{(t_r)_w\,(Q_w)}{0.12\,d^2}$$

$$h_o + h_w = \frac{(10)(5{,}000 + 3{,}000)}{0.12\,d^2} = \frac{667{,}000}{d^2}$$

5. Compute combinations of d, and $h_o + h_w$ for diameters greater than d_{min} (Table 5-1).

Table 5-1
Vertical Three-Phase Separator Capacity
Diameter vs. Length for Retention Time Constraint
$(t_r)_o = (t_r)_w$ = 10 minutes

d in.	$h_o + h_w$ in.	L_{ss} ft	(12) L_{ss}/d
84	94.5	18.2	2.6
90	82.3	17.7	2.4
96	72.3	17.4	2.2
102	64.1	17.2	2.0

6. Compute seam-to-seam length (Table 5-1) as the larger of:

$$L_{ss} = \frac{h_o + h_w + 76}{12} \text{ or } \frac{h_o + h_w + d + 40}{12}$$

7. Compute slenderness ratio (12 L_{ss}/d). Choices in the range of 1.5 to 3 are common (Table 5-1).
8. If necessary repeat steps 6 through 10 for various retention times and graph as was done two-phase separators.
9. Choose a reasonable size. A 90-in. × 15-ft or a 96-in. × 12-ft 6-in. size would be a reasonable choice.

Example 5-2: Sizing a Horizontal Three-Phase Separator

Given:

Q_o = 5,000 bopd
Q_w = 3,000 bwpd
Q_g = 5 MMscfd
P = 100 psia
T = 90°F
Oil = 30° API
$(S.G.)_w$ = 1.07
S_g = 0.6
$(t_r)_o = (t_r)_w$ = 10 min.
Oil viscosity = 10 cp

Solution:

1. Calculate difference in specific gravities.

$$°API = \frac{141.5}{(S.G.)_o} - 131.5$$

$$(S.G.)_o = \frac{141.5}{30 + 131.5}$$

$$\Delta S.G. = 1.07 - 0.876 = 0.194$$

2. Check for gas separation. See Chapter 4 procedure.

$dL_{eff} = 102$

Table 5-2
Horizontal Three-Phase Separator
Diameter vs. Length for Gas Capacity Restraint

d in.	L_{eff} ft
60	1.7
72	1.4
84	1.2
96	1.1

3. Calculate combinations of d and L_{eff} for gas separation (Table 5-2). Because of the low values for L_{eff}, gas capacity will not govern.

4. Calculate maximum oil pad thickness.

$$(h_o)_{max} = 0.00128 \frac{(t_r)_o\,(\Delta S.G.)\, d_m^2}{\mu}$$

$$= 0.00128 \frac{(10)\,(0.194)\,(500)^2}{10}$$

$$= 62.1$$

5. Calculate maximum diameter for oil pad thickness constraint.

$$(h_o)_{max} = 0.00128 \frac{(t_r)_o\,(\Delta S.G.)\, d_m^2}{\mu}$$

$$= 0.00128 \frac{(10)\,(0.194)\,(500)^2}{10}$$

$$= 0.1875$$

From Figure 5-8:

$\beta = 0.257$

$$d_{max} = \frac{(h_o)_{max}}{\beta}$$

$$= \frac{62.1}{0.257}$$

$d_{max} = 24.16$ in.

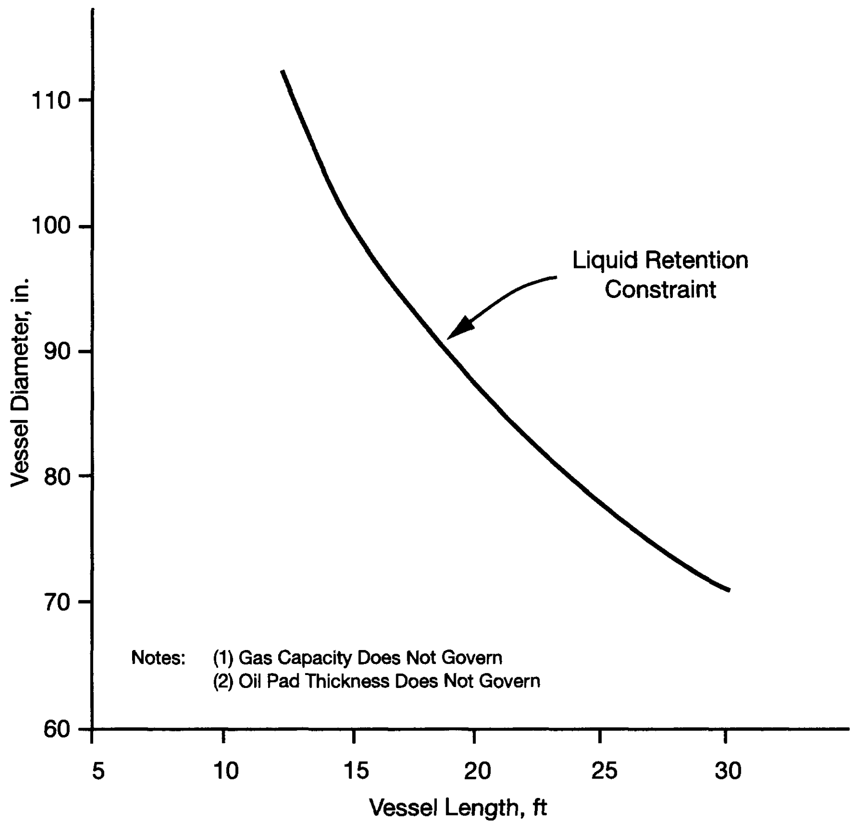

Figure 5-9. Horizontal three-phase separator example.

Table 5-3
Horizontal Three-Phase Separator Capacity
Diameter vs. Length for Liquid Retention Time Constraint
$(t_r)_o = (t_r)_w = 10$ minutes

d in.	L_{eff} ft	L_{ss} ft	(12) L_{ss}/d
60	31.6	42.1	8.4
72	21.9	29.2	4.9
84	16.1	21.5	3.1
96	12.3	16.4	2.1
108	9.7	13.0	1.4

6. Liquid retention constraint:

$$\begin{aligned} d^2 L_{eff} &= 1.42\,[Q_w\,(t_r)_w + Q_o\,(t_r)_o] \\ &= (1.42)(10)(8{,}000) \\ &= 113{,}600 \end{aligned}$$

7. Compute combinations of d and L_{eff} (Table 5-3).

8. Compute seam-to-seam length (Table 5-3).

$$L_{ss} = \frac{L_{eff}}{0.75} \text{ or } L_{ss} = L_{eff} + \frac{d}{12}$$

9. Compute slenderness ratio ($12\ L_{ss}/d$). Choices in the range of 3 to 5 are common.

10. Graph results and choose a reasonable size that does not violate gas capacity restraint or oil pad thickness restraint. Possible choices from Figure 5-9 are 90 in. × 20 ft, 96 in. × 17 ft 6 in., and 102 in. × 15 ft.

CHAPTER

6

Crude Oil Treating Systems*

INTRODUCTION

Removing water from crude oil often requires additional processing beyond gravitational separation. In selecting a treating system, several factors should be considered to determine the most desirable methods of treating the crude oil to contract requirements. Some of these factors are:

1. Tightness of the emulsion.
2. Specific gravity of the oil and produced water.
3. Corrosiveness of the crude oil, produced water, and casinghead gas.
4. Scaling tendencies of the produced water.
5. Quantity of fluid to be treated and percent water in the fluid.
6. Paraffin-forming tendencies of the crude oil.
7. Desirable operating pressures for equipment.
8. Availability of a sales outlet and value of the casinghead gas produced.

A common method for separating this "water-in-oil" emulsion is to heat the stream. Increasing the temperature of the two immiscible liquids

*Reviewed for the 1998 edition by Matthew A. McKinstry of Paragon Engineering Services, Inc.

deactivates the emulsifying agent, allowing the dispersed water droplets to collide. As the droplets collide they grow in size and begin to settle. If designed properly, the water will settle to the bottom of the treating vessel due to differences in specific gravity.

The process of coalescence requires that the water droplets have adequate time to contact each other. It also assumes that the buoyant forces on the coalesced droplets are sufficient to enable these droplets to settle to the bottom of the treating vessel. Consequently, design considerations should necessarily include temperature, time, viscous properties of oil that inhibit settling, and the physical dimensions of the vessel, which determine the velocity at which settling must occur.

Laboratory analysis, in conjunction with field experience, should be the basis for specifying the configuration of treating vessels. The purpose of this chapter is to present a rational alternative for those instances when laboratory data do not exist or, if it is desirable, to extrapolate field experience.

EMULSION TREATING THEORY

Forming Emulsions

For an emulsion to exist there must be two mutually immiscible liquids, an emulsifying agent, and sufficient agitation to disperse the discontinuous phase into the continuous phase. In oil production, oil and water are the two mutually immiscible liquids. An emulsifying agent in the form of small solid particles, paraffins, asphaltenes, etc., is almost always present in the formation fluids, and sufficient agitation always occurs as fluid makes its way into the well bore, up the tubing, and through the surface choke.

The difficulty of separating the emulsified water from the oil depends on the "stability" of the emulsion. The stability of an emulsion is dependent on several factors:

1. The difference in density between the water and oil phases.
2. The size of dispersed water particles.
3. Viscosity.
4. Interfacial tension.
5. The presence and concentration of emulsifying agents.

The difference in density is one of the factors that determines the rate at which water droplets drop through the continuous oil phase. The greater the difference in density, the more quickly water droplets will settle from the oil phase. The water particle size also affects the rate at which water particles move through the oil phase. The larger the particle, the faster it will settle out of the oil phase. The water particle size in an emulsion is dependent upon the degree of agitation that the emulsion is subject to before treating. Flow through pumps, chokes, valves, and other surface equipment will decrease water particle sizes.

Viscosity plays two primary roles. First, as viscosity increases, more agitation is required to shear water particles down to a smaller average size in the oil phase. Therefore, the size of water particles that must be removed to meet water cut specifications for a given treating system increases as viscosity increases. Second, as viscosity increases, the rate at which water particles move through the oil phase decreases, resulting in less coalescence and increased difficulty in treating.

When no emulsifier is present, the interfacial tension between oil and water is high. When interfacial tension is high, water particles coalesce easily upon contact. When emulsifying agents are present, however, they decrease the interfacial tension and obstruct the coalescence of water particles.

The above factors determine the "stability" of emulsions. Some stable emulsions may take weeks or months to separate if left alone in a tank with no treating. Other unstable emulsions may separate into relatively clean oil and water phases in just a matter of minutes.

Normal oil field emulsions consist of an oil continuous or external phase, and a water dispersed or internal phase. In some isolated cases, where there are high water cuts, it is possible to form reverse emulsion with water as the continuous phase and oil droplets the internal phase. Complex emulsions have been reported in low gravity, viscous crudes. These mixed emulsions contain a water external phase and have an internal water phase in the dispersed oil. The vast majority of oil treating systems deal with normal emulsions and that is what is discussed in this chapter.

Figure 6-1 shows a normal emulsion. The small water droplets exist within the oil continuous phase. Figure 6-2 shows a close up of a "skin" of emulsifying agent surrounding a water drop, and Figure 6-3 shows two drops touching, but being prevented from coalescing due to the film of emulsifying agent around each drop.

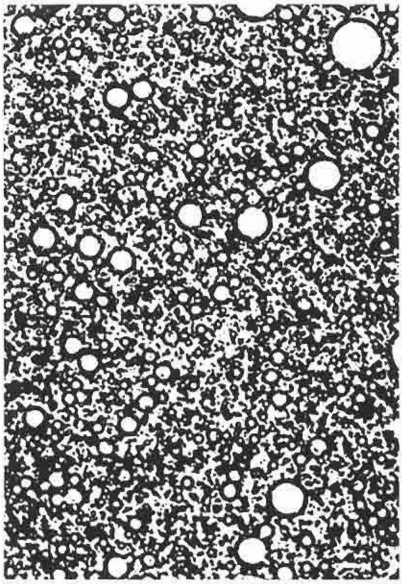

Figure 6-1. Photomicrograph of normal emulsion.

Figure 6-2. Photomicrograph of emulsifying agent surrounding a water droplet.

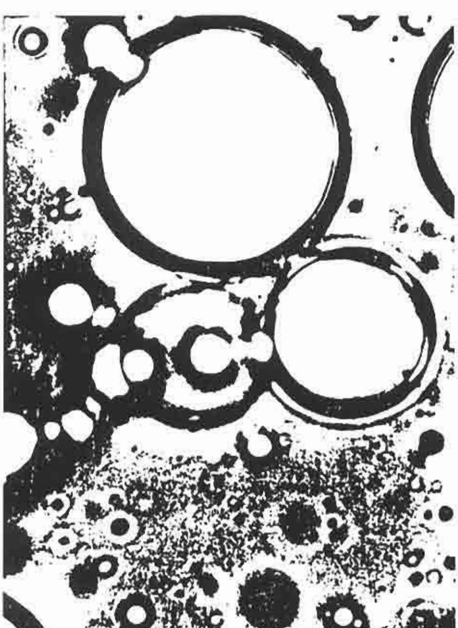

Figure 6-3. Photomicrograph of emulsifying agent preventing two droplets from coalescing.

Emulsifying Agent

When thinking about emulsion stability, it may be helpful to realize that in a pure oil and pure water mixture, without an emulsifying agent, no amount of agitation will create an emulsion. If the pure oil and water are mixed and placed in a container, they quickly separate. The natural state is for the immiscible liquids to establish the least contact or smallest surface area. The water dispersed in the oil forms spherical drops. Smaller drops will coalesce into larger drops and this will create a smaller interface area for a given volume. If no emulsifier is present, the droplets will eventually settle to the bottom causing the smallest interface area. This type of mixture is a true "dispersion."

An emulsifying agent has a surface active behavior. Some element in the emulsifier has a preference for the oil, and other elements are more attracted to the water. An emulsifier tends to be insoluble in one of the liquid phases. It thus concentrates at the interface. There are several ways an emulsifier changes a dispersion into an emulsion. The action of the emulsifier can be visualized as one or more of the following:

1. It decreases the interfacial tension of the water droplet, thus causing smaller droplets to form. The smaller droplets take longer to coalesce into larger droplets, which can settle quickly.
2. If forms a viscous coating on the droplets that keeps them from coalescing into larger droplets when they collide. Since coalescence is prevented, it takes longer for the small droplets created by agitation to settle out.
3. The emulsifiers may be polar molecules, which align themselves in such a manner as to cause an electrical charge on the surface of the droplets. Since like electrical charges repel, two droplets must collide with sufficient force to overcome this repulsion before coalescence can occur.

Naturally-occurring surface active materials normally found in crude oil serve as emulsifiers. Paraffins, resins, organic acids, metallic salts, colloidal silts and clay, and asphaltenes (a general term for material with chemical compositions containing sulfur, nitrogen, and oxygen) are common emulsifiers in oil fields. Workover fluids and drilling mud are also sources of emulsifying agents.

The type and amount of emulsifying agent has an immediate effect on the emulsion's stability. It has been shown that the temperature history of

the emulsion is also important as it effects the formation of paraffins and asphaltenes. The speed of migration of the emulsifying agent to the oil/water interface and the behavior in terms of the strength of the interface bond are important factors. An emulsion treated soon after agitation or the creation of paraffins and asphaltenes can be less stable and easier to process if the migration of the emulsifier is incomplete. An aged emulsion may become more difficult to treat. Normally, the lower the crude viscosity and lighter the crude the more rapid the aging process. Therefore, early treatment may be a lesser factor in treating low-viscosity, high-API-gravity crudes.

Demulsifiers

Chemical demulsifiers sold under various trade names, such as Tretolite™, Visco™, and Breaxit™, are highly useful in resolving emulsions. Demulsifiers act to neutralize the effect of emulsifying agents. Typically, they are surface active agents and thus their excessive use can decrease the surface tension of water droplets and actually create more stable emulsions.

There are four important actions required of a demulsifier:

1. Strong attraction to the oil-water interface.
2. Flocculation.
3. Coalescence.
4. Solid wetting.

When these actions are present, they promote the separation of oil and water. The demulsifier must have the ability to migrate rapidly through the oil phase to the droplet interface, where it must compete with the more concentrated emulsifying agent. The demulsifier must also have an attraction for droplets with a similar condition. In this way large clusters of droplets gather which, under a microscope, appear like bunches of fish eggs. The oil will take on a bright appearance since small droplets are no longer present to scatter the light rays. At this point the emulsifier film is still continuous. If the emulsifier is weak, the flocculation force may be enough to cause coalescence. This is not true in most cases and the demulsifier must therefore neutralize the emulsifier and promote a rupture of the droplet interface film. This is the opener that causes coalescence. With the emulsion in a flocculated condition the film rupture results in rapid growth of water-drop size.

The manner in which the demulsifier neutralizes the emulsifier depends upon the type of emulsifiers. Iron sulfides, clays, and drilling muds can be water wet causing them to leave the interface and be diffused into the water droplet. Paraffins and asphaltenes could be dissolved or altered to make their films less viscous so they will flow out of the way on collision or could be made oil wet so they will be dispersed in the oil.

It would be unusual if one chemical structure could produce all four desirable actions. A blend of compounds is therefore used to achieve the right balance of activity.

The demulsifier selection should be made with the process system in mind. If the treating process is a settling tank, a relatively slow-acting compound can be applied with good results. On the other hand, if the system is a chemelectric process where some of the flocculation and coalescing action is accomplished by an electric field, there is need for a quick-acting compound, but not one that must complete the droplet-building action.

Emulsion-breaking chemicals are most commonly tested with bottle tests, which involve mixing various chemicals with samples of the emulsion and observing the results. Such tests are effective in eliminating some chemicals and selecting those that appear to be more efficient. Bottle tests also provide an estimate of the amount of chemical required. Bottle tests should be performed on a representative sample as soon as the sample is obtained because of the possible detrimental effects of aging. These tests should also be performed at conditions that are as close to field treating conditions as possible. Synthetic water should not be used in place of produced water in bottle tests because the produced water may have very different properties, and it may contain impurities that are not present in the synthetic water.

While candidate chemicals and approximate dosages can be determined in bottle tests, the dynamic nature of the actual flowing system requires that several candidates be field-tested. In actual conditions, the emulsion undergoes shearing through control valves, coalescence in flow through pipes, and changes to the emulsion that occur inside the treating vessel as a result of inlet diverters, water wash sections, etc. Static bottle tests cannot model these dynamic conditions.

As field conditions change, the chemical requirements can change. If the process is modified, e.g., very low rates on electrostatic units, the chemical requirement can change. Seasonal changes bring paraffin-

induced emulsion problems. Workovers contribute to solids content, which alters emulsion stability. So, no matter how satisfactory a demulsifier is at one point in time, it cannot be assumed that it will always be satisfactory over the life of the field.

GRAVITY SEPARATION

Most oil-treating equipment relies on gravity to separate water droplets from the oil continuous phase, because water droplets are heavier than the volume of oil they displace. However, gravity is resisted by a drag force caused by the droplets' downward movement through the oil. When the two forces are equal, a constant velocity is reached, which can be computed from Stokes' Law as:

$$V_t = \frac{1.78 \times 10^{-6} \, (\Delta S.G.) \, (d_m)^2}{\mu} \qquad (6-1)$$

where V_t = downward velocity of the water droplet relative to the oil continuous phase, ft/s
d_m = diameter of the water droplet, micron
$\Delta S.G.$ = difference in specific gravity between the oil and water
μ = dynamic viscosity of the oil continuous phase, centipoise (cp)

(Stokes' Law was derived in Chapter 4.)

Several conclusions can be drawn from Stokes' Law:

1. The larger the size of a water droplet, the larger the square of its diameter, and thus, the greater its downward velocity. That is, the bigger the droplet size, the less time it takes for the droplet to settle to the bottom of the vessel and thus the easier it is to treat the oil.
2. The greater the difference in density between the water droplet and the oil phase, the greater the downward velocity. That is, the lighter the crude, the easier it is to treat the oil. If the crude gravity is 10°API and the water fresh, the settling velocity is zero, as there is no gravity difference.
3. The higher the temperature, the lower the viscosity of the oil, and thus the greater the downward velocity. That is, it is easier to treat the oil at high temperatures than at low temperatures (assuming a small effect on gravity difference due to increased temperature).

Coalescence

The process of coalescence in oil treating systems is time dependent. In dispersions of two immiscible liquids, immediate coalescence seldom occurs when two droplets collide. If the droplet pair is exposed to turbulent pressure fluctuations, and the kinetic energy of the oscillations induced in the coalescing droplet pair is larger than the energy of adhesion between them, the contact will be broken before coalescence is completed.

Experiments with deep layer gravity settlers indicate that the time to "grow" a droplet size due to coalescence can be estimated by the following equation:

$$t = \frac{\pi}{6}\left(\frac{d^j - (d_o)^j}{\phi K_S}\right) \tag{6-2}$$

where d_o = initial droplet size
d = final droplet size
ϕ = volume fraction of the dispersed phase
K_s = empirical parameter for the particular system
j = an empirical parameter always larger than 3 and dependent on the probability that the droplets will "bounce" apart before coalescence occurs

When the energy of oscillations is very low so that "bouncing" of droplets approaches zero, j approaches 3. Assuming a value of 4, the minimum time required to obtain a desired particle diameter can be expressed:

$$t = \frac{\pi}{6}\left(\frac{d^4 - (d_o)^4}{\phi K_S}\right) \tag{6-3}$$

Assuming d_o is small relative to the droplet size we wish to "grow" by coalescence in our gravity settler, Equation 6-3 can be approximated:

$$t = \frac{d^4}{2\phi K_S} \tag{6-4}$$

The following qualitative conclusions for coalescence in a gravity settler can be drawn from this relationship:

1. A doubling of residence time increases the maximum size drop grown in a gravity settler less than 19 percent. If the value of "j" is greater than 4 the growth in droplet diameter will be even slower.

2. The more dilute the dispersed phase, the greater the residence time needed to "grow" a given particle size. That is, coalescence occurs more rapidly in concentrated dispersions. This is the reason that oil is "water washed" by entering the treating vessel below the oil-water interface in most gunbarrels and treaters. Flocculation and coalescence therefore occur most effectively at the interface zone between oil and water.

Viscosity

Laboratory testing of a particular oil at various temperatures is the most reliable method of determining how an oil behaves. ASTM D 341 outlines a procedure where the viscosity is measured at two different temperatures and then either through a computation or special graph paper the viscosity at any other temperature can be obtained. Figure 6-4 shows a portion of one of these graphs.

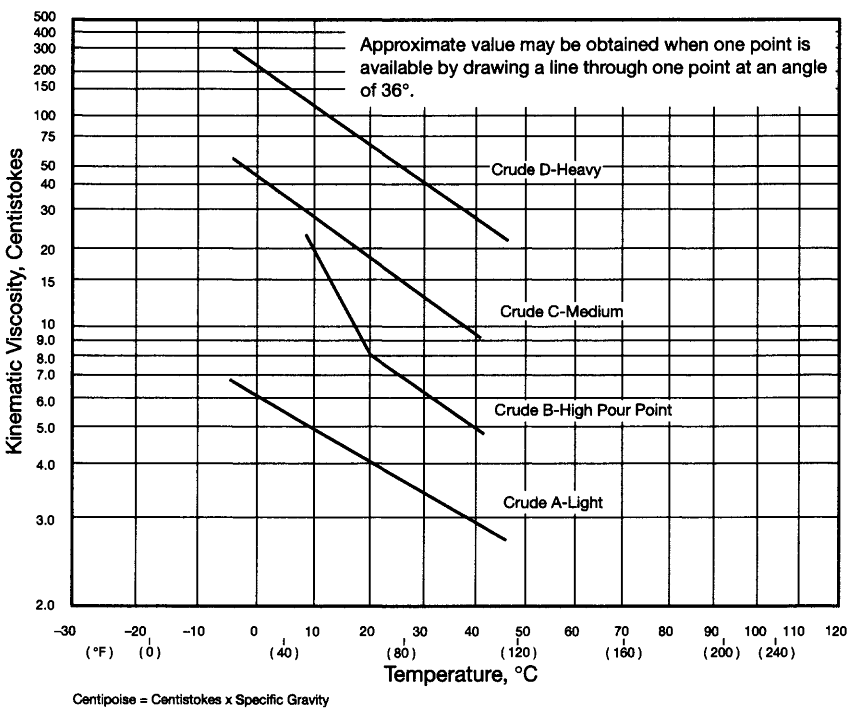

Figure 6-4. Viscosity-temperature graph for crude oils (*courtesy of ASTM D-341*).

As a rule, with crude of 30° API and higher the viscosity is so low that normally it may be difficult to find any information on file regarding a specific crude viscosity. Between 30° API and 11° API, the viscosity becomes more important, until in some cases it is impossible to process very low gravity crudes without a diluent to reduce the viscosity. The use of a diluent is not unusual for crude oil below 14° API.

With virtually any crude oil the viscosity change with temperatures can be an excellent guide to minimum crude processing temperatures. An ASTM chart of the viscosity versus temperature is useful to detect the paraffin formation or cloud point of the crude as shown in Figure 6-4. This normally establishes a minimum temperature for the treating process. There are examples of 30° API crude and higher that have pour points of 80° to 90°F. Crude oils of this type are common in the Uinta and Green River Basins of the United States as well as S.E. Asia.

In the absence of any laboratory data, Chapter 3 discusses correlations that can be used to estimate crude viscosity given its gravity and temperature.

Temperature Effects

Adding heat to the incoming oil/water stream is the traditional method of separating the phases. The addition of heat reduces the viscosity of the oil phase allowing more rapid settling velocities in accordance with Equation 6-1. It also has the effect of dissolving the small crystals of paraffin and asphaltenes and thus neutralizing their effect as potential emulsifiers. Treating temperatures normally range from 100–160°F. In treating of heavy crudes the temperature may be as high as 300°F.

Adding heat can cause a significant loss of the lower boiling point hydrocarbons (light ends). This results in a "shrinkage" of the oil, or loss of volume. The molecules leaving the oil phase may be vented or compressed and sold with the gas. Even if they are sold with the gas, there will be probably be a net *loss* in income realized by converting liquid volume into gas volume. Figure 6-5 shows the amount of shrinkage that may be expected.

Increasing the temperature at which treating occurs also has the disadvantage of making the crude oil that is recovered in the storage tank heavier and thus decreasing its value. Because the light ends are boiled off, the remaining liquid has a lower API gravity. Figure 6-6 shows the API gravity loss for a typical crude oil.

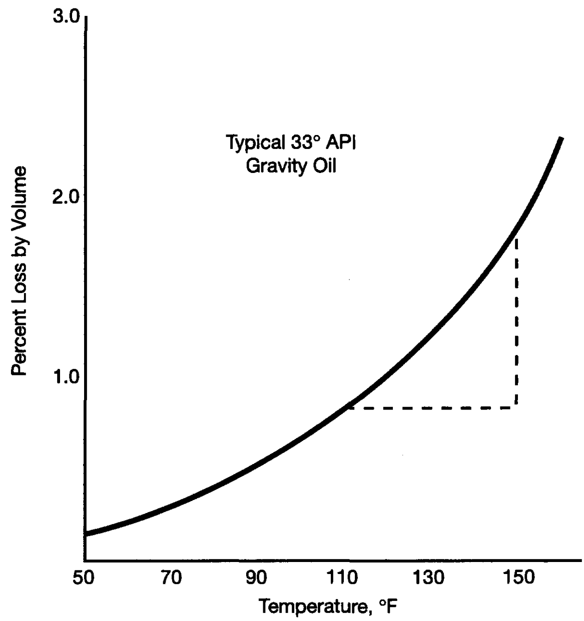

Figure 6-5. Percent loss by volume vs. temperature.

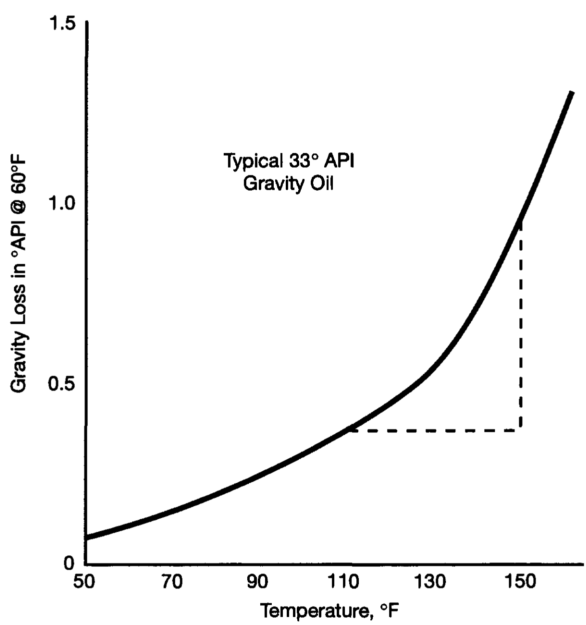

Figure 6-6. API gravity loss vs. temperature.

Increasing the temperature may lower the specific gravity at treater operating pressure of both the oil to be treated and the water that must be separated from it. However, depending on the properties of the crude it may either increase or decrease the difference in specific gravity as shown in Figure 6-7. In most cases, if the treating temperature is less than 200°F the change in S.G. with temperature can be neglected.

Finally, it takes fuel to provide heat and the cost of fuel must be considered. Thus, while heat may be needed to adequately treat the crude, the less heat that is used, the better. Table 6-1 illustrates the overall economic effect of treating temperature for a lease that produces 21,000 bopd of a 29° API crude.

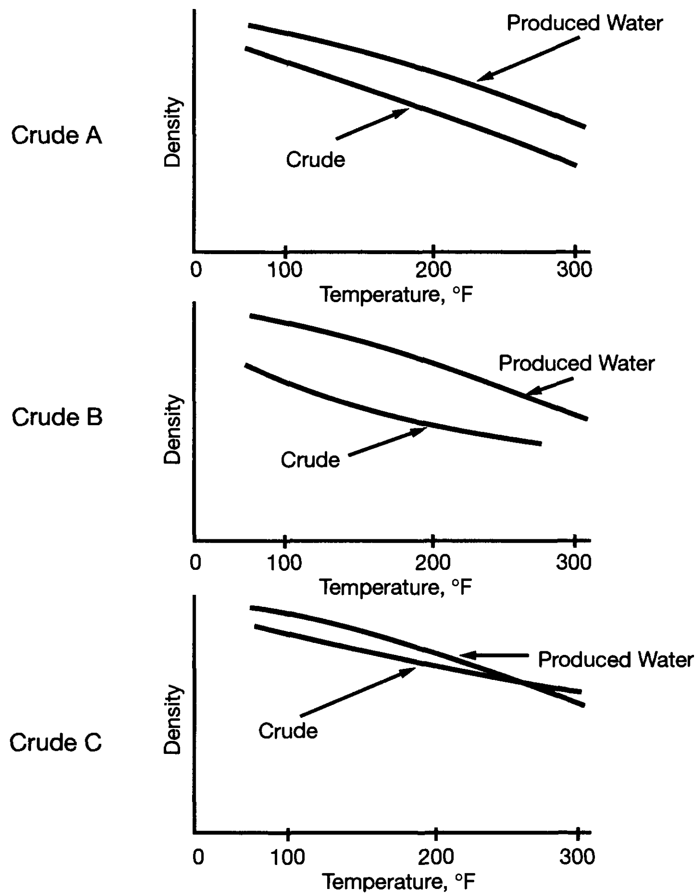

Figure 6-7. Relationship of specific gravity with temperature for three crude oils.

Table 6-1
Economic Effect of Treating at a Higher Temperature

Increase in NGL Value:

Component	Volume for 120°F Treater Temperature	Volume for 100°F Treater Temperature	Difference	Price* Per Unit	Change In NGL Revenue
Methane	163 mcfh	162 mcfh	1 mcfh	$2.44	$2.44/hr
Ethane	1,835 gal/hr	1,802 gal/hr	33 gal/hr	0.232	7.66/hr
Propane	1,653 gal/hr	1,527 gal/hr	126 gal/hr	0.439	55.31/hr
Butane	1,086 gal/hr	930 gal/hr	156 gal/hr	0.672	104.83/hr
Pentane+	1,251 gal/hr	968 gal/hr	283 gal/hr	0.749	211.97/hr
					$382.21/hr
				$/Day	
		Total NGL Revenue Gain		$9,173.04	
		Net Value to Producer		$3,057.68	
Volume Shrinkage					
317 bopd × $19.52/bbl				(6,187.84)	
API Gravity Loss:					
20,931 bpd × $0.15 /bbl × 7/10 (API Change) =				(2,197.75)	
Fuel Cost					
125 Mcfd × $2.44/Mcf =				(306.00)	
TOTAL LEASE REVENUE LOSS:				$(5,633.91)	

Source: Heiman, V. L. et. al.: "Maximize Revenue by Analyzing Crude Oil Treating" Society of Petroleum Engineers of AIME, SPE 12206 (October 1983).
** February 1983 prices.*

The gas liberated when crude oil is heated may create a problem in the treating equipment if the equipment is not properly designed. In vertical heater-treaters and gunbarrels the gas rises through the coalescing section. If much gas is liberated, it can create enough turbulence and disturbance to inhibit coalescence. Perhaps more important is the fact that the small gas bubbles have an attraction for surface active material and hence for the water droplets. The bubbles thus have a tendency to keep the water droplets from settling and may even cause them to carry-over to the oil outlet.

The usual oil field horizontal heater-treater tends to overcome the gas liberation problem by coming to equilibrium in the heating section before introducing the emulsion to the settling-coalescing section. Some large crude processing systems use a fluid-packed, pump-through system that keeps the crude well above the bubble point. Top-mount degassing separators above electrostatic coalescers have been used in some installations.

If properly and prudently done, heating an emulsion can greatly benefit water separation. However, if a satisfactory rate of water removal can be achieved at the minimum temperature delivered into a process, there may be no reason to suffer the economic penalties associated with adding heat.

Heat Input Equations

The heat input and thus the fuel required for treating depend on the temperature rise, amount of water in the oil, and flow rate. It requires about twice as much energy to heat water as it does to heat oil. For this reason, it is beneficial to separate any free water from the emulsion to be treated with either a free-water knockout located upstream of the treater or an inlet free-water knockout system in the treater itself.

Assuming that the free water has been separated from the emulsion, the water remaining is less than 10% of the oil, and the treater is insulated to minimize heat losses, the required heat input can be determined from:

$$q = 15\, Q_o\, \Delta T\, [0.5\, (S.G.)_o + 0.1] \qquad (6-5)$$

where q = heat input, Btu/hr
Q_o = oil flow rate, bopd
ΔT = increase in temperature, °F
$S.G._o$ = specific gravity of oil relative to water

Derivation of Equation 6-5

General heat transfer equation is expressed by:

$$q = W\, c\, \Delta T$$

where q = heat (Btu/hr)
W = flow rate (lb/hr)
c = specific heat (Btu/lb-°F) (approximately 0.5 for oil and 1.0 for water)
ΔT = temperature increase (°F)

Since water weighs 350 lb/bbl,

$$W = \frac{350}{24}\, (S.G.)_l\, Q_l$$

where $S.G._l$ = specific gravity of the liquid
Q_l = liquid flow rate (bpd)

The total energy required is determined from:

$$q = q_o + q_w + q_{lost}$$

where q = total energy required to heat the stream
q_o = energy required to heat the oil
$= [(350/24)\ (S.G.)_o\ Q_o]\ (0.5)\ \Delta T$
q_w = energy required to heat the water
$= [(350/24)\ (S.G.)_w\ Q_w]\ (1.0)\ \Delta T$
q_{lost} = energy lost to surroundings, assume 10% of total heat input (q)

Substituting:

$$q = (350/24)[(S.G.)_o\ Q_o\ (0.5) + (S.G.)_w\ Q_w]\ \Delta T + (0.1)\ q$$

Assume 10% water and specific gravity water = 1:

$$q = 16\ Q_o\ \Delta T\ [0.5\ (S.G.)_o + (0.1)]$$

Water Droplet Size and Retention Time

The droplet diameter is the most important single parameter to control to aid in water settling since this term is squared in the settling equation. A small increase in diameter will create a much larger increase in settling rate.

It would be extremely rare to have laboratory data of droplet coalescence for a given system. Qualitatively we would expect droplet size to increase with retention time in the coalescing section, and with heat input, which excites the system leading to more collisions of small droplets. Droplet size could be expected to decrease with oil viscosity, which inhibits the movement of the particles and decreases the force of the collision.

The coalescence equation indicates that the oil-water interface zone is where nearly all of the coalescence occurs. Except for providing some minimal time for initial coalescence to occur, increasing retention time in a crude oil treating system may not be very cost effective. Consequently, in most systems, if one does not design the hydraulics properly at this zone, the opportunity can be lost to grow water droplets of sufficient size to settle in the vessel.

Coalescing Media

It is possible to use a coalescing media to promote coalescence of the water droplets. These media provide large surface area upon which water droplets can collect. The most common coalescing media is wood shav-

ings or excelsior, which is referred to as a "hay section." The wood excelsior is tightly packed to create an obstruction to the flow of the small water droplets and promote random collision of these droplets for coalescence. When the droplets are large enough, they will fall out of the flow stream by gravity.

It is possible that the use of a hay section will allow lower treating temperatures. However, these media have a tendency to clog with time and are difficult to remove. They are no longer in common use.

Electrostatic Coalescers

Coalescing of the small water drops dispersed in the crude can be accomplished by subjecting the water-in-oil emulsion to a high-voltage electrical field. When a non-conductive liquid (oil) containing a dispersed conductive liquid (water) is subjected to an electrostatic field, the conductive particles or droplets are caused to combine by one of three physical phenomena:

1. The droplets become polarized and tend to align themselves with the lines of electric force. In so doing, the positive and negative poles of the droplets are brought adjacent to each other. Electrical attraction brings the droplets together and causes them to coalesce.
2. Droplets are attracted to an electrode due to an induced charge. In an A-C field, due to inertia, small droplets vibrate over a larger distance than larger droplets promoting coalescence. In a D-C field the droplets tend to collect on the electrodes forming larger and larger drops until eventually they fall by gravity.
3. The electric field tends to distort and thus weaken the film of emulsifier surrounding the water droplets. Water droplets dispersed in oil and subjected to a sinusoidal alternating-current field will be elongated along the lines of force during the first half cycle. As they are relaxed during the low-voltage portion, the surface tension will pull the droplets back toward the spherical shape. The same effect is obtained in the next half of the alternating cycle. The weakened film is thus more easily broken when droplets collide, making coalescence more likely.

Whatever the actual mechanism, the electric field causes the droplets to move about rapidly in random directions, which greatly increases the chances of collision with another droplet. When droplets collide with the proper velocity, coalescence occurs.

The attraction between water droplets in an electric field is given by:

$$F = \frac{K_s \, \varepsilon^2 \, (d_m)^6}{S^4} \text{ (with } S \geq d_m\text{)} \tag{6-6}$$

where F = attractive force between droplets
K_s = constant for system
ε = voltage gradient
d_m = diameter of droplets
S = distance between droplets

This equation indicates that the greater the voltage gradient the greater for the forces causing coalescence. However, experimental data show that at some gradient the water droplet can be pulled apart and a strong emulsion can be developed. For this reason electrostatic treaters are normally equipped with a mechanism for adjusting the gradient in the field.

TREATING EQUIPMENT

Vertical Treaters

The most commonly used single-well lease treater is the vertical treater as shown in Figure 6-8. Flow enters the top of the treater into a gas separation section. Care must be exercised to size this section so that it has adequate dimensions to separate the gas from the inlet flow. If the treater is located downstream of a separator, this chamber can be very small. The gas separation section should have an inlet diverter and a mist extractor.

The liquids flow through a downcomer to the base of the treater, which serves as a free-water knockout section. If the treater is located downstream of a free-water knockout, the bottom section can be very small. If the total wellstream is to be treated this section should be sized for 3 to 5 minutes retention time for both the oil and the water to allow the free water to settle out. This will minimize the amount of fuel gas needed to heat the liquid stream rising through the heating section. The end of the downcomer should be slightly below the oil water interface to "water wash" the oil being treated. This will assist in the coalescence of water droplets in the oil.

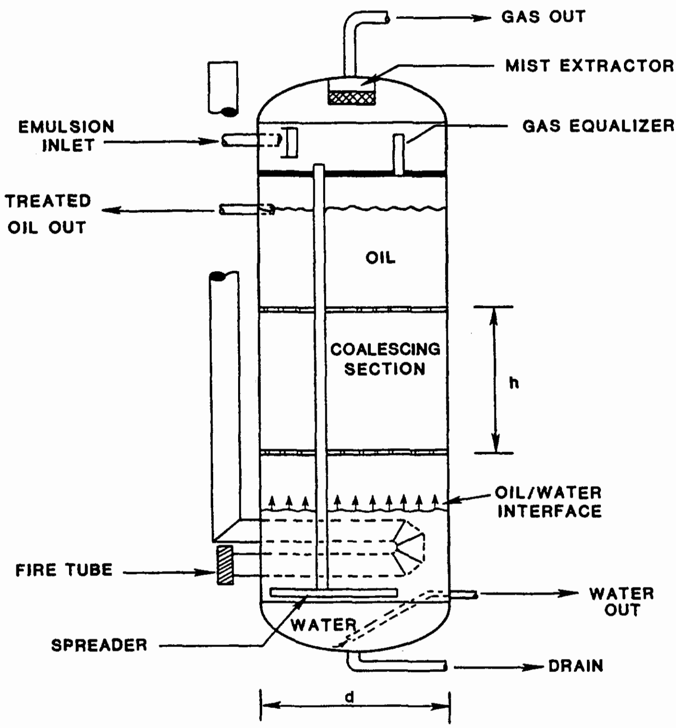

Figure 6-8. Vertical treater schematic.

The oil and emulsion rises over the heater fire-tubes to a coalescing section where sufficient retention time is provided to allow the small water particles in the oil continuous phase to coalesce and settle to the bottom.

Treated oil flows out the oil outlet. Any gas, flashed from the oil due to heating, flows through the equalizing line to the gas space above. Oil level is maintained by pneumatic or lever operated dump valves. Oil-water interface is controlled by an interface controller, or an adjustable external water leg.

The detailed design of the treater, including the design of internals (many features of which are patented) should be the responsibility of the equipment supplier.

Figure 6-9 shows a "gunbarrel" tank, which is a vertical flow treater in an atmospheric tank. Typically, gunbarrels have a gas separating chamber or "boot" on top where gas is separated and vented, and a downcomer. Because gunbarrels tend to be of larger diameter than vertical heater-

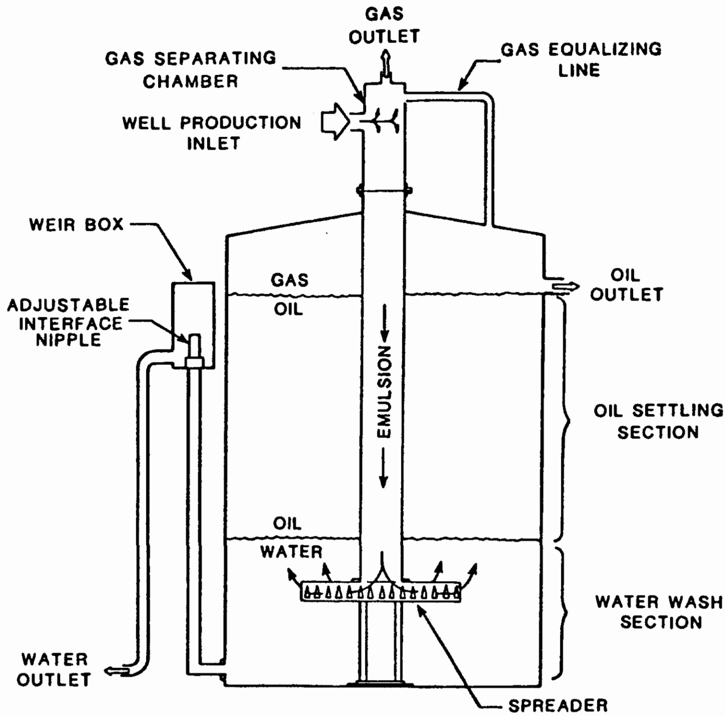

Figure 6-9. Typical gunbarrel settling tank with internal flume.

treaters, many have elaborate spreader systems to try and create uniform (i.e., plug) upward flow of the emulsion to take maximum advantage of the entire cross section. Most gunbarrels are unheated, though it is possible to provide heat by heating the incoming stream external to the tank, installing heating coils in the tank, or circulating the water to an external or "jug" heater in a closed loop. It is preferable to heat the inlet so that more gas is liberated in the boot, although this means that fuel will be used in heating any free water in the inlet.

Gunbarrels are most often used in older, small flow rate, onshore facilities. In recent times vertical heater-treaters have become so inexpensive that they have replaced gunbarrels in single well installations. On larger installations onshore in warm weather areas gunbarrels are still commonly used. In areas that have a winter season it tends to be too expensive to keep the large volume of oil at a high enough temperature to combat potential pour point problems.

Horizontal Treaters

For most multi-well situations horizontal treaters are normally required. Figure 6-10 shows a typical design of a horizontal treater.

Flow enters the front section of the treater where gas is flashed. The liquid falls around the outside to the vicinity of the oil-water interface where the liquid is "water washed" and the free water is separated. Oil and emulsion rise past the fire tubes and are skimmed into the oil surge chamber. The oil-water interface in the inlet section of the vessel is controlled by an interface level controller, which operates a dump valve for the free water.

The oil and emulsion flow through a spreader into the back or coalescing section of the vessel, which is fluid packed. The spreader distributes the flow evenly throughout the length of this section. Treated oil is collected at the top through a collection device sized to maintain uniform vertical flow of the oil. Coalescing water droplets fall countercurrent to the rising oil continuous phase. The oil-water interface is maintained by a level controller and dump valve for this section of the vessel.

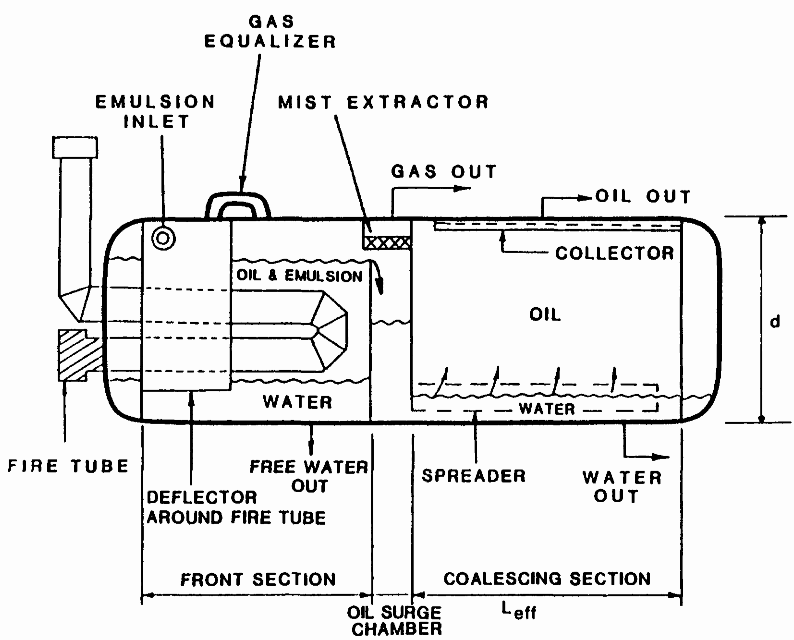

Figure 6-10. Horizontal heater-treater schematic.

A level control in the oil surge chamber operates a dump valve on the oil outlet line regulating the flow of oil out the top of the vessel to maintain a fluid packed condition.

The inlet section must be sized to handle settling of the free water and heating of the oil. The coalescing section must be sized to provide adequate retention time for coalescence to occur and to allow the coalescing water droplets to settle downward countercurrent to the upward flow of the oil.

Electrostatic Treaters

Some treaters use an electrode section. Figure 6-11 illustrates a typical design of a horizontal electrostatic treater. The flow path in an electrostatic treater is the same as a horizontal treater. The only difference is that an AC and/or DC electrostatic field is used to promote coalescence of the water droplets.

Procedures for designing electrostatic coalescers have not been published. Since coalescence of droplets in an electric field is so dependent

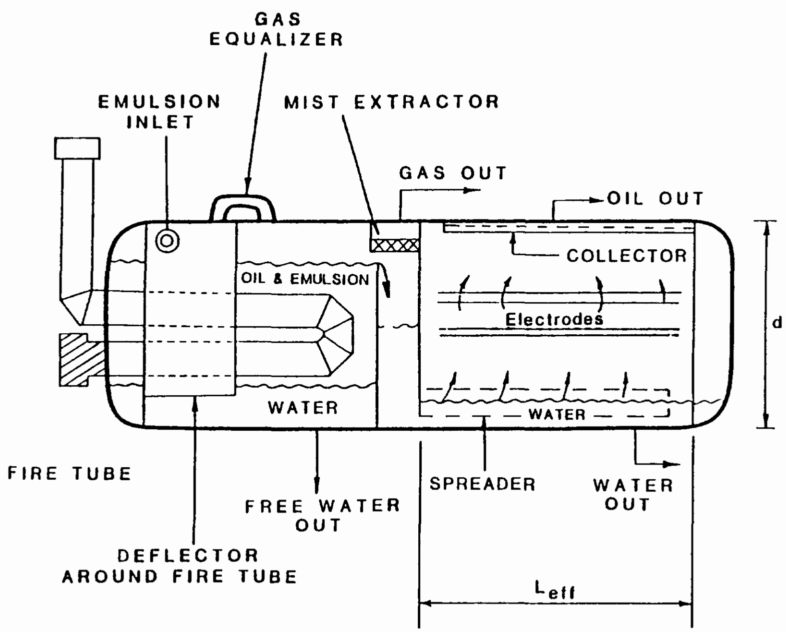

Figure 6-11. Horizontal electrostatic treater schematic.

on the characteristics of the particular emulsion to be treated, it is unlikely that a general relationship of water droplet size to use in the settling equations can be developed. Field experience tends to indicate that electrostatic treaters are efficient at reducing water content in the crude below the 0.5 to 1.0% basic sediment and water (BS&W) level. This makes them particularly attractive for desalting applications. However, for normal crude treating, where 0.5 to 1.0% BS&W is acceptable, it is recommended that they be sized as heater-treaters. By trial and error after installation, the electric grids may be able to allow treating to occur at lower temperatures.

EQUIPMENT SIZING AND THEORY

Settling Equations

The specific gravity difference between the dispersed water droplets and the oil should result in the water "sinking" to the bottom of the treatment vessel.

Since the oil continuous phase is flowing vertically upward in both vertical and horizontal treaters previously described, the downward velocity of the water droplet must be sufficient to overcome the velocity of the oil traveling upward through the treater. By setting the settling velocity equal to the oil velocity the following general sizing equations can be derived:

Horizontal Vessels

$$d\, L_{eff} = 438 \frac{Q_o\, \mu}{(\Delta S.G.)\, d_m{}^2} \qquad (6-7)$$

Vertical Vessels

$$d = 81.8 \left[\frac{Q_o\, \mu}{(\Delta S.G.)\, d_m^2} \right]^{1/2} \qquad (6-8)$$

Gunbarrels

$$d = 81.8 \left[\frac{F Q_o\, \mu}{(\Delta S.G.)\, d_m^2} \right]^{1/2} \qquad (6-9)$$

where d = diameter of vessel, in.
Q_o = oil flow rate, bopd
μ = oil viscosity, cp
L_{eff} = length of coalescing section, ft
$\Delta S.G.$ = difference in specific gravity between oil and water (relative to water)
d_m = diameter of water droplet, microns
F = short-circuiting factor (1.0 for $d < 48$ inches, and greater than 1.0 for $d > 48$ inches)

Derivation of Equation 6-7

V_t and V_o are in ft/s, d_m in micron, μ in cp

$$V_t = V_o$$

$$V_t = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)\,{d_m}^2}{\mu}$$

Q is in ft/s, A in ft^2, Q_o in bpd, d in inches

$$V_o = \frac{Q}{A}$$

$$Q = 6.49 \times 10^{-5}\,Q_o$$

$$A = \frac{d}{12} \times L_{eff}$$

$$V_o = 7.79 \times 10^{-4}\,\frac{Q_o}{d\,L_{eff}}$$

$$d\,L_{eff} = 438\,\frac{Q_o\,\mu}{(\Delta S.G.)\,{d_m}^2}$$

Derivation of Equations 6-8 and 6-9

V_t and V_o are in ft/s, d_m in micron, μ in cp

$$V_t = V_o$$

$$V_t = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)\,{d_m}^2}{\mu}$$

Q is in ft/s, A in ft^2, Q_o in bpd, d in inches

$$V_o = \frac{Q}{A}$$

$$V_o = 0.0119 \frac{Q_o}{d^2}$$

$$d^2 = 6{,}690 \frac{Q_o \mu}{(\Delta S.G.)\, d_m{}^2}$$

$$d = 81.8 \left[\frac{Q_o\, \mu}{(\Delta S.G.)\, d_m{}^2} \right]^{1/2}$$

Note that the height of the coalescing section for a vertical treater does not enter the settling equation. The cross sectional area of flow for the upward velocity of the oil is a function of the diameter of the vessel alone. In a horizontal vessel the cross sectional area for flow for the upward velocity of the oil is a function of the diameter times the length of the coalescing section.

The sizing equation for gunbarrels includes a short-circuiting factor (F). This factor accounts for imperfect liquid distribution across the entire cross-section of the treating vessel or tank and is a function of the flow conditions in the vessel. The larger the retention time, the larger the short-circuiting factor. It may be necessary to apply a short-circuiting factor for large vertical treaters as well.

Retention Time Equations

The oil must be held at temperature for a specific period of time to enable de-emulsifying the water-in-oil emulsion. This information is best determined in the laboratory but, in the absence of such data, 20 to 30 minutes is a good starting point.

Depending on the specific properties of the stream to be treated, geometry required to provide a certain retention time may be larger or smaller than the geometry required to satisfy the settling equation. The geometry of the vessel is determined by the larger of the two criteria. The equations for retention time are as follows:

Horizontal Vessels

$$d^2 L_{eff} = \frac{Q_o\, (t_r)_o}{1.05} \qquad (6-10)$$

Vertical Vessels

$$d^2\,h = \frac{(t_r)_o\,Q_o}{0.12} \qquad (6-11)$$

Gunbarrels

$$d^2h = \frac{F(t_r)_o\,Q_o}{0.12} \qquad (6-12)$$

where t_r = retention time, min
Q_o = oil flow, bopd
h = height of the coalescing section, in.
F = short-circuiting factor (1.0 for $d < 48$ inches, and greater than 1.0 for $d > 48$ inches)

Derivation of Equation 6-10

t is in s, vol in ft^3, Q in ft^3/s, D in ft, d in inches, L_{eff} in ft

$$t = \frac{Vol}{Q}$$

Assuming only 75% of the cross-sectional area is effective

$$Vol = 0.75\left(\frac{\pi\,D^2\,L_{eff}}{4}\right) = \frac{(0.75)\,\pi\,d^2\,L_{eff}}{(4)\,(144)}$$

$Q = 6.49 \times 10^{-5}\,Q_o$

$d^2L_{eff} = 0.0159\,Q_o t$

$(t_r)_o$ is in minutes

$t = 60(t_r)_o$

$$d^2\,L_{eff} = \frac{Q_o\,(t_r)_o}{1.05}$$

Equations 6-10 and 6-12 are derived in the same manner as the retention time equation for horizontal separators.

Water Droplet Size

In order to develop a treater design procedure, the water droplet size to be used in the settling equation to achieve a given outlet water cut must be determined. As previously mentioned, it would be extremely rare to

have laboratory data of the droplet size distribution for a given emulsion as it enters the coalescing section of the treater. Qualitatively, we would expect the minimum droplet size that must be removed for a given water cut to (1) increase with retention time in the coalescing section, (2) increase with temperature, which tends to excite the system, leading to more collisions of small droplets, and (3) increase with oil viscosity, which tends to inhibit the formation of small droplets from shearing that occurs in the system.

We have seen that, after an initial period, increasing the retention time has a small impact on the rate of growth of particles. Thus, for practically sized treaters with retention times of 10 to 30 minutes, retention time would not be expected to be a determinant variable. Intuitively, one would expect viscosity to have a much greater effect on coalescence than temperature.

Assuming that the minimum required size of droplets that must be settled is a function only of oil viscosity, equations have been developed correlating this droplet size and oil viscosity [1]. The authors used data from three conventional treaters operating with 1% water cuts. Water droplet sizes were back-calculated using Equation 6-7. The calculated droplet sizes were correlated with oil viscosity, and the following equation resulted:

$$d_{m1\%} = 200\,\mu^{0.25} \qquad \mu_o < 80 \text{ cp} \tag{6-13}$$

where $d_{m1\%}$ = diameter of water droplet to be settled from the oil to achieve 1% water cut, microns

μ = viscosity of the oil phase, cp

Using the same procedure, the following correlation for droplet size was developed for electrostatic treaters:

$$d_{m1\%} = 170\,\mu^{0.4} \qquad 3 \text{ cp} < \mu_o < 80 \text{ cp} \tag{6-14}$$

For viscosities below 3 cp, Equation 6-8 should be used. The two equations intersect at 3 cp, and electrostatic treaters would not be expected to operate less efficiently in this range. Additionally, the data from which the electrostatic treater droplet size correlation was developed did not include oil viscosities less than 7 cp.

The same authors also investigated the effect of water cut on minimum droplet size. Data from both conventional and electrostatic treaters over a range of water cuts were used to back-calculate an imputed droplet size as a function of water cut, resulting in the following equation:

$$\frac{d_m}{d_{m1\%}} = W_c^{0.33} \tag{6-15}$$

where d_m = diameter of water droplet to be settled from the oil to achieve a given water cut (W_c), microns
W_c = water cut, percent

As the volume of a sphere is proportional to the diameter cubed, Equation 6-10 indicates that the water cut is proportional to the droplet diameter cubed.

It must be stressed that the above equations should be used only in the absence of other data and experience. These proposed relationships are based only on limited experimental data.

An approximate sizing relationship, derived from Equations 6-13 and 6-14, is given in Figure 6-12 in terms of the flow rate of emulsion (given in bpd) flowing vertically through a horizontal cross-sectional area of

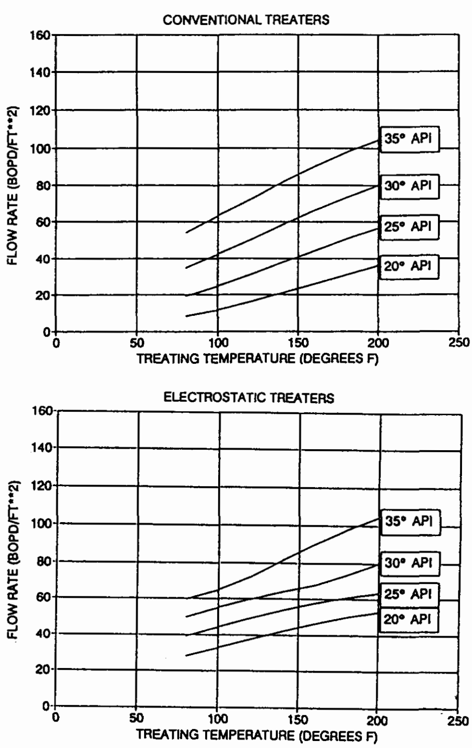

Figure 6-12. Flow rate vs. treating temperature for conventional and electrostatic treaters.

one square foot. For a horizontal treater with vertical flow through the coalescing section, the flow area can be approximated as the diameter of the vessel times the length of the coalescing section.

DESIGN PROCEDURE

In specifying the size of a treater, it is necessary to determine the diameter (d), length or height of the coalescing section (L_{eff} or h), and treating temperature or fire-tube rating. As we have seen, these variables are interdependent, and it is not possible to arrive at a unique solution for each. The design engineer must trade the cost of increased geometry against the savings from reducing the treating temperature.

The equations previously presented provide tools for arriving at this trade-off. However, because of the empirical nature of some of the underlying assumptions engineering judgment must be utilized in selecting the size of treater to use. The general procedure is outlined as follows:

1. Choose a treating temperature.
2. Determine oil viscosity at treating temperature.
3. Determine the diameter of the water droplet that must be removed from the oil at treating temperature from Equation 6-13 or 6-14, along with Equation 6-15.
4. Determine the treater geometry necessary to satisfy settling criteria from Equation 6-7 for horizontal vessels, Equation 6-8 for vertical vessels, or Equation 6-9 for gunbarrels.
5. Check the geometry to assure it provides sufficient retention time as indicated by Equation 6-10 for horizontal vessels, Equation 6-11 for vertical vessels, or Equation 6-12 for gunbarrels.
6. Repeat the procedure for different assumed treating temperatures.

This procedure allows the production facility engineer to choose the major sizing parameters of heater-treaters when little or no laboratory data are available. This procedure does not give the overall dimensions of the treater, which must include inlet gas separation and free-water knockout sections. However, it does provide a method for specifying a fire-tube capacity and a minimum size for the coalescing section (where the treating actually occurs), and provides the engineer with the tools necessary to evaluate specific vendor proposals.

EXAMPLES

Example 6-1: Sizing a Horizontal Treater

Given: Oil gravity = 30°API, 0.875 S.G.
Oil flow rate = 5,000 bpd
Inlet oil temperature = 80°F
Water S.G. = 1.04
Inlet BS&W = 10%
Outlet BS&W = 1%

Solution:

1. **Settling Equation.** Investigate treating at 80°F, 100°F, 120°F.

Treating Temperature	80°F	100°F	120°F
ΔS.G.	0.165	0.165	0.165
μ_o	40	15	9
d_m	503	394	346
d L_{eff}	2,098	1,283	998

2. **Retention Time Equation.** Plot computations of d and L_{eff} with retention times less than 20 minutes.

   $d^2 L_{eff} = 20\ (5{,}000)/1.05 = 95{,}238$

   The shaded area of Figure 6-13 represents combinations of d and L_{eff} with t_r less than 20 minutes.

3. **Heat Required**

   $q = (16)\ (5{,}000)\ (\Delta T)\ [\ (0.5)\ (0.876) + 0.1]$

   $q = 43{,}040\ (\Delta T)$

   Substituting treating temperature values of 80°F, 100°F, and 120°F and substituting initial oil temperature value of 80°F will yield values of heat required of 0, 0.86, and 1.72 MMBtu/h.

4. **Selection.** Choose any combination of d and L_{eff} that is not in the shaded area. Read corresponding treating temperature.

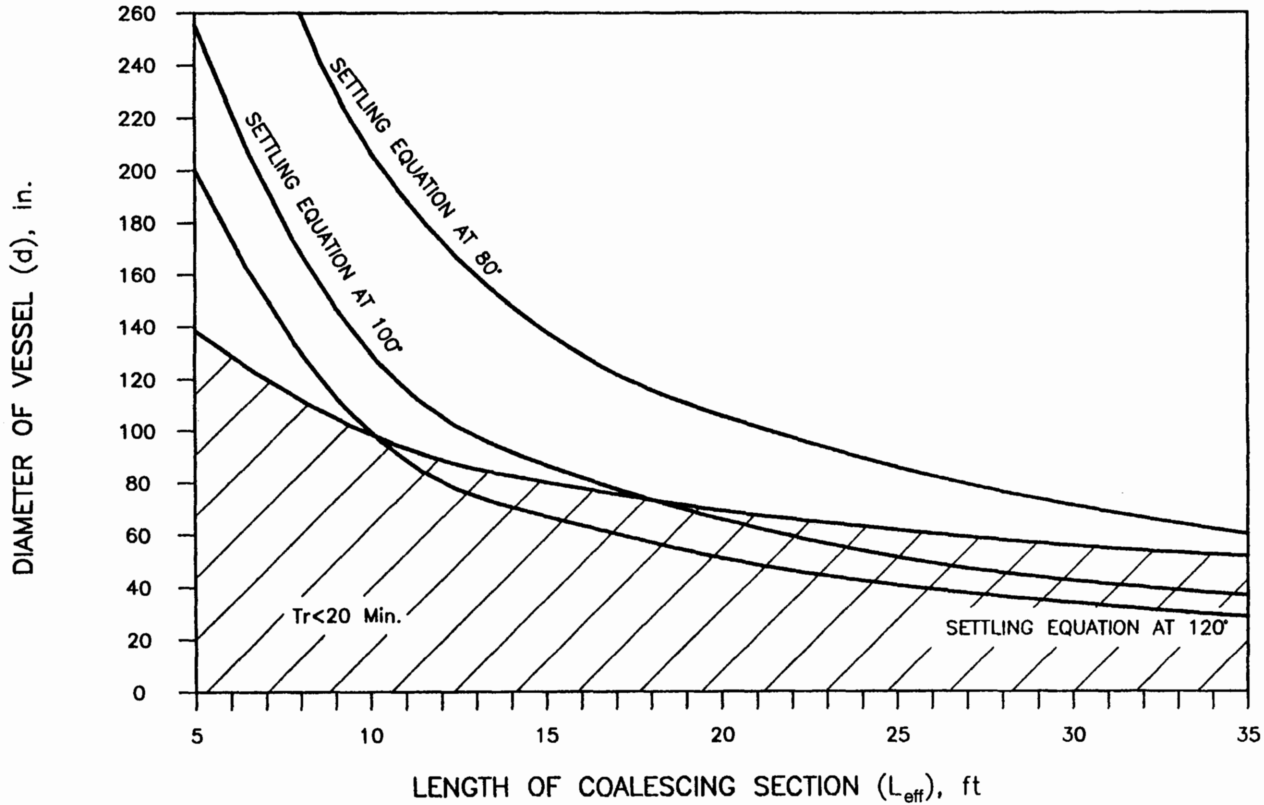

Figure 6-13. Horizontal treater example.

Example solutions are:

Treating Temperature, °F	d, in.	L_{eff}, ft	Heat Required, MMBtu/h
80°F	144	15	0.00
	120	18	
	96	22	
100°F	96	14	0.86
	72	20	
120°F	96	10	1.72
	72	20	

An economical solution would be a 72-inch-diameter treater with a 20-foot coalescing section and a 0.86-MMBtu/h firetube capacity. Given the nature of empirical design procedures, crude could possibly be treated at 80°F. The additional firetube capacity will allow a temperature of 100°F if required by field conditions.

Example 6-2: Sizing a Vertical Treater

Given: Oil gravity = 40°API, 0.875 S.G.

Oil flow rate = 2,000 bpd

Inlet oil temperature = 90°F

Water S.G. = 1.04

Inlet BS&W = 10%

Outlet BS&W = 1%

Solution:

1. **Settling Equation.** Investigate treating at 90°F, 100°F, 120°F.

Treating Temperature	90°F	100°F	120°F
ΔS.G.	0.215	0.215	0.215
μ_o	7.0	5.1	3.3
d_m	325	301	270
$d\ L_{eff}$	64	59	53

2. **Retention Time.** Plot computations of d and h with retention times less than 20 minutes.

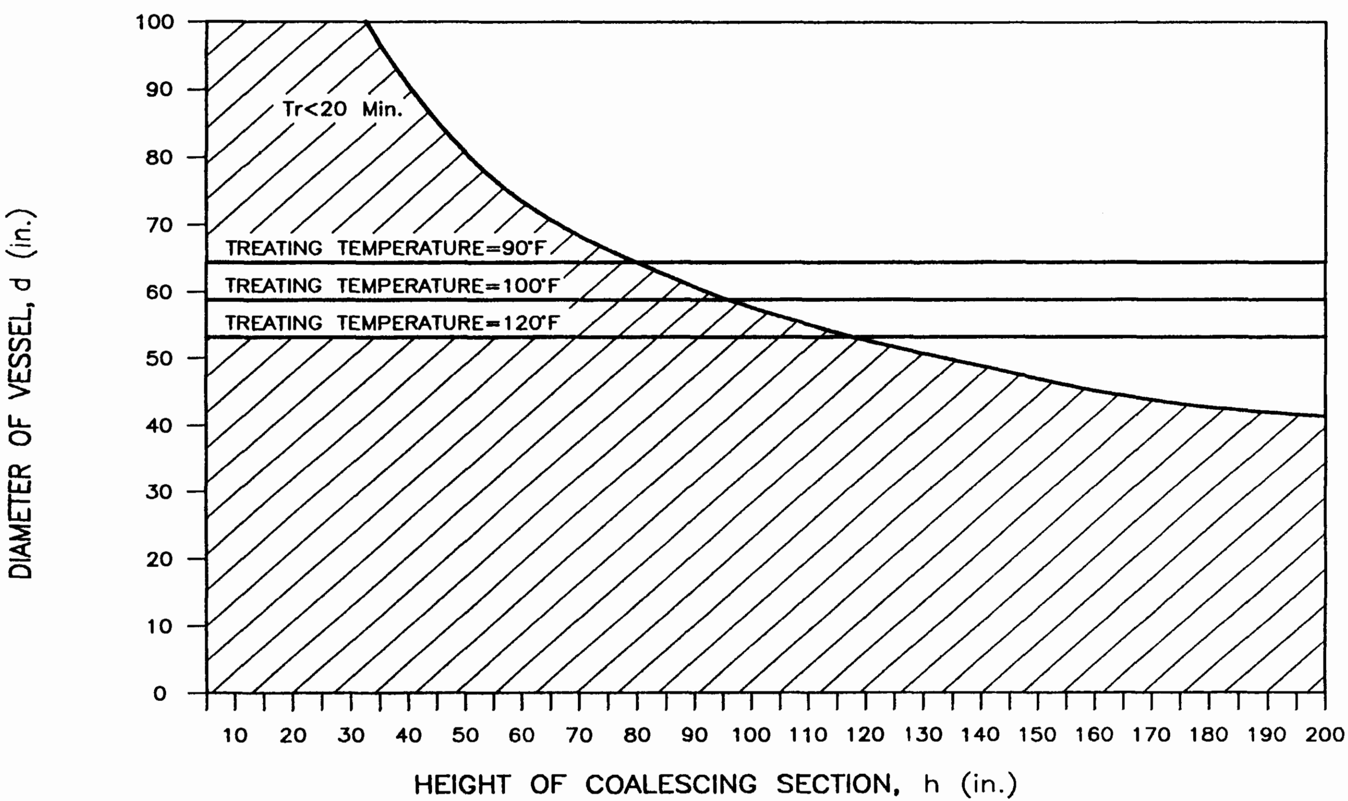

Figure 6-14. Vertical treater example.

$d^2 h = (20) (2{,}000)/0.12 = 333{,}333$

The shaded area of Figure 6-14 represents combinations of d and h with t_r less than 20 minutes.

3. **Heat Required**

$q = (16) (2{,}000) (\Delta T) [(0.5) (0.825) + 0.1]$

$q = 16{,}400 (\Delta T)$

4. **Selection.** Choose any combination of d and h that is not in the shaded area. Read the corresponding treating temperature.

Example solutions are:

Treating Temperature, °F	d, in.	h, in.	Heat Required, MMBtu/h
120°F	53	120	0.49
100°F	59	100	0.16
90°F	64	90	0

An economical solution would be a 60-inch-diameter treater with a 100-inch-high coalescing section and a 0.16-MMBtu/h firetube capacity. In actual service, crude may not require heating at all. The firetube capacity will allow a treating temperature of 100°F if required by field conditions.

REFERENCES

1. Thro, M. E., and Arnold, K. E., "Water Droplet Size Determination for Improved Oil Treater Sizing," Society of Petroleum Engineers 69th Annual Technical Conference and Exhibition, New Orleans, LA, 1994.

CHAPTER

7

*Produced-Water Treating Systems**

INTRODUCTION

In producing operations it is often necessary to handle wastewater that may include water produced with crude oil, rain water, and washdown water. The water must be separated from the crude oil and disposed of in a manner that does not violate established environmental regulations. In offshore areas where discharge to the sea is allowed, the governing regulatory body specifies the maximum hydrocarbon content in the water that may be discharged overboard. The range is currently 15 mg/l to 50 mg/l depending on the specific location. In most onshore locations the water cannot be disposed of on the surface, due to possible salt contamination, and must be injected into an acceptable disposal formation or disposed of by evaporation. In either case it will probably be necessary to treat the produced water to lower its hydrocarbon content below that normally obtained from free-water knockouts and oil treaters. The purpose of this chapter is to present the engineer with a procedure for selecting the appropriate type of equipment for treating oil from produced water and to provide the theoretical equations and empirical rules necessary to size

*Reviewed for the 1998 edition by Michael E. Whitworth of Paragon Engineering Services, Inc.

the equipment. When this design procedure is followed, the engineer will be able to develop a process flowsheet, determine equipment sizes, and evaluate vendor proposals for any wastewater treating system.

SYSTEM DESCRIPTION

Table 7-1 lists the various methods employed in produced-water treating systems and the types of equipment that employ each method. Figure 7-1 shows a typical produced-water treating system configuration. Produced water will always have some form of primary treating prior to disposal. This could be a skim tank, skim vessel, CPI, or crossflow separator. All of these devices employ gravity separation techniques. Depending upon the severity of the treating problem, secondary treating utilizing a CPI, crossflow separator, or a flotation unit may be required. Liquid-liquid hydrocyclones are often used either in a single stage or with a downstream skim vessel or flotation unit.

Table 7-1
Produced-Water Treating Equipment

Method	Equipment Type	Approximate Minimum Drop Size Removal Capabilities (Microns)
Gravity Separation	Skimmer Tanks and Vessels API Separators Disposal Piles Skim Piles	100–150
Plate Coalescence	Parallel Plate Interceptors Corrugated Plate Interceptors Cross-Flow Separators Mixed-Flow Separators	30–50
Enhanced Coalescence	Precipitators Filter/Coalescers Free-Flow Turbulent Coalescers	10–15
Gas Flotation	Dissolved Gas Hydraulic Dispersed Gas Mechanical Dispersed Gas	15–20
Enhanced Gravity Separation	Hydrocyclones Centrifuges	5–15
Filtration	Multi-Media Membrane	1+

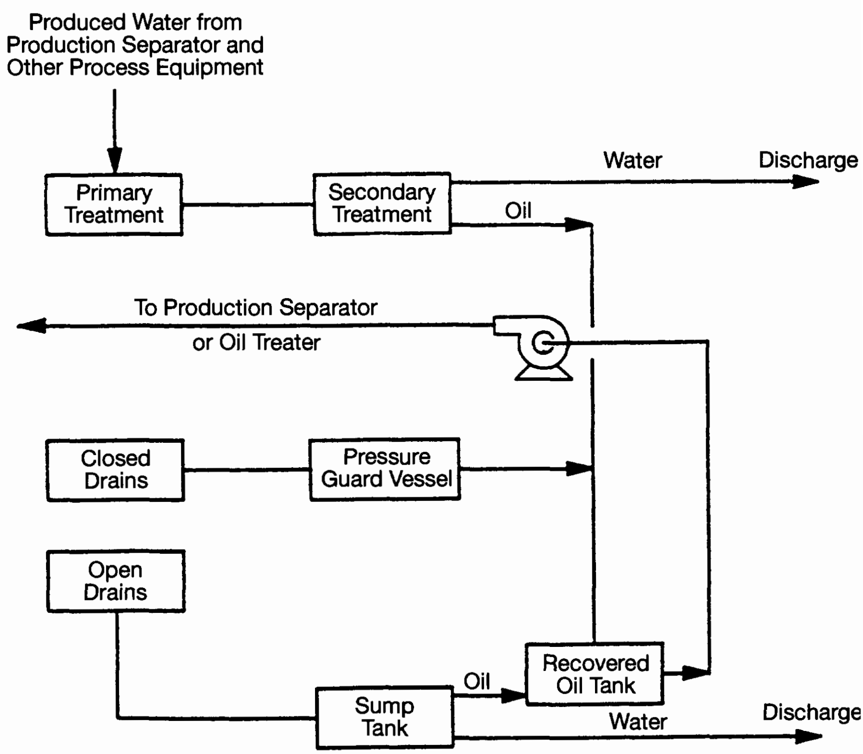

Figure 7-1. Typical produced-water treating system.

Offshore, produced water can be piped directly overboard after treating, or it can be routed through a disposal pile or a skim pile. Deck drains must be treated for removal of "free" oil. This is normally done in a skim vessel called a sump tank. Water from the sump tank is either combined with the produced water or routed separately for disposal overboard.

Onshore, the water will normally be reinjected in the formation or be pumped into a disposal well.

For safety considerations, closed drains, if they exist in the process, should never be tied into atmospheric drains and should be routed to a pressure vessel prior to entering an atmospheric tank or pile. This could be done in a skim vessel, crossflow separator or CPI in a pressure vessel. The latter two could be used where it is desirable to separate sand from the system.

THEORY

The function of all water treating equipment is to cause the oil droplets that exist in the water continuous phase to separate from the water phase so they can then be removed. In gravity separation units, the difference in specific gravity causes the oil to float to the surface of the water. The oil droplets are subjected to continuous dispersion and coalescence during the trip up the wellbore through the surface chokes, flowlines, control valves and the process equipment. When energy is put into the system at a high rate the drops are dispersed to smaller sizes. When the energy input rate is low, small droplets collide and join together in the process of coalescence.

Gravity Separation

Most commonly used water treating equipment items rely on the forces of gravity to separate the oil droplets from the water continuous phase. The oil droplets, being lighter than the volume of water they displace, have a buoyant force exerted upon them. This is resisted by a drag force caused by their vertical movement through the water. When the two forces are equal, a constant velocity is reached, which can be computed from Stokes' Law as:

$$V_t = \frac{1.78 \times 10^{-6} \, (\Delta S.G.)\,(d_m)^2}{\mu} \qquad (7-1)$$

where V_t = terminal settling velocity, ft/s
d_m = diameter of the oil droplet, micron
$\Delta S.G.$ = difference in specific gravity of oil and water relative to water
μ = viscosity of the water continuous phase, cp

Several conclusions can be drawn from this simple equation:

1. The larger the size of an oil droplet, the larger the square of its diameter, and, thus, the greater its vertical velocity. That is, the bigger the droplet size, the less time it takes for the droplet to rise to a collection surface and thus the easier it is to treat the water.
2. The greater the difference in density between the oil droplet and the water phase, the greater the vertical velocity. That is, the lighter the crude, the easier it is to treat the water.

3. The higher the temperature, the lower the viscosity of the water, and thus the greater the vertical velocity. That is, it is easier to treat the water at high temperatures than at low temperatures.

Dispersion

An oscillating droplet of oil becomes unstable when the kinetic energy is sufficient to make up for the difference in the surface energy between the single droplet and the two smaller droplets formed from it. At the same time that this process is occurring, the motion of the smaller oil particles is causing coalescence to occur. Therefore, it should be possible to define statistically a maximum droplet size for a given energy input per unit mass and time at which the rate of coalescence equals the rate of dispersion.

One relationship for the maximum particle size that can exist at equilibrium was proposed by Hinze as follows:

$$d_{max} = 432\left(\frac{t_r}{\Delta P}\right)^{2/5}\left(\frac{\sigma}{\rho_w}\right)^{3/5} \qquad (7-2)$$

where d_{max} = diameter of droplet above which size only 5% of the oil volume is contained, micron
σ = surface tension, dynes/cm
ρ_w = density, g/cm^3
ΔP = pressure drop, psi
t_r = retention time, minutes

It can be seen that the greater the pressure drop and thus the shear forces that the fluid experiences in a given period of time while flowing through the treating system, the smaller the maximum oil droplet diameter will be. That is, large pressure drops that occur in small distances through chokes, control valves, desanders, etc., result in smaller droplets.

The dispersion process is theoretically not instantaneous. However, it appears from field experience to occur very rapidly. For design purposes, it could be assumed that whenever large pressure drops occur, all droplets larger than d_{max} will instantaneously disperse. This is, of course, a conservative approximation.

Coalescence

The process of coalescence in water treating systems is more time dependent than the process of dispersion. In dispersions of two immisci-

ble liquids, immediate coalescence seldom occurs when two droplets collide. If the droplet pair is exposed to turbulent pressure fluctuations, and the kinetic energy of the oscillations induced in the droplet pair is larger than the energy of adhesion between them, the contact will be broken before coalescence is completed.

It has been shown in the previous chapter that the time to "grow" a droplet size due to coalescence in a gravity settler is proportional to the diameter of the droplet to some power greater than three and inversely proportional to the concentration of the oil phase. From this it can be concluded that after an initial period of coalescence in a settler, additional retention time has a rapidly diminishing ability to cause coalescence and to capture oil droplets.

Flotation

The process of flotation improves the separation of the oil droplets from the water continuous phase. This goal is accomplished by increasing the difference in density between the two fluids by attaching gas bubbles to the oil droplet. The flotation process will decrease vessel retention time, thereby decreasing the separating vessel size required to allow a specified droplet size to float to the surface or decreasing the size of oil droplet that can be captured by a separation vessel of specified size.

TREATING EQUIPMENT

Settling Tanks and Skimmer Vessels

The simplest form of primary treating equipment is a settling (skim) tank or vessel. These items are normally designed to provide long residence times during which coalescence and gravity separation can occur. If the desired outlet oil concentration is known, the theoretical dimensions of the vessel can be determined. Unlike the case of separation, with skim vessels one cannot ignore the effects of vibration, turbulence, short-circuiting, etc. American Petroleum Institute (API) Publication 421, *Management of Water Discharges: Design and Operation of Oil-Water Separators,* uses short-circuit factors as high as 1.75 and is the basis upon which many of the sizing formulas in this chapter were derived.

Skimmers can be either vertical or horizontal in configuration. In vertical skimmers the oil droplets must rise upward countercurrent to the

downward flow of the water. Some vertical skimmers have inlet spreaders and outlet collectors to help even the distribution of the flow, as shown in Figure 7-2. The inlet directs the flow below the oil-water interface. Small amounts of gas liberated from the water help to "float" the oil droplets. In the quiet zone between the spreader and the water collector, some coalescence can occur and the buoyancy of the oil droplets causes them to rise counter to the water flow. Oil will be collected and skimmed off the surface.

The thickness of the oil pad depends on the relative heights of the oil weir and the water leg, and the difference in specific gravity of the two liquids. Often, an interface lever controller is used in place of the water leg.

In horizontal skimmers the oil droplets rise perpendicular to the flow of the water, as shown in Figure 7-3. The inlet enters below the oil pad. The water then turns and flows horizontally for most of the length of the vessel. Baffles could be installed to straighten the flow. Oil droplets coa-

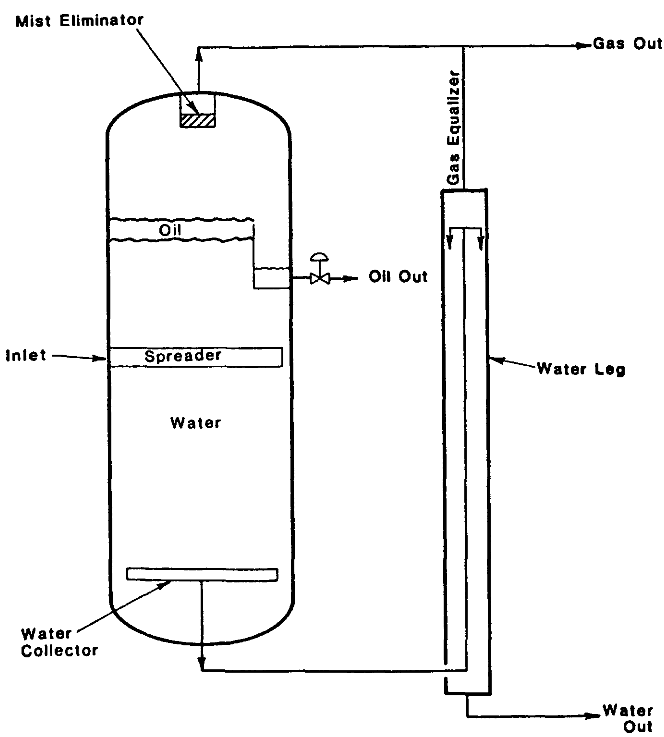

Figure 7-2. Vertical skimmer schematic.

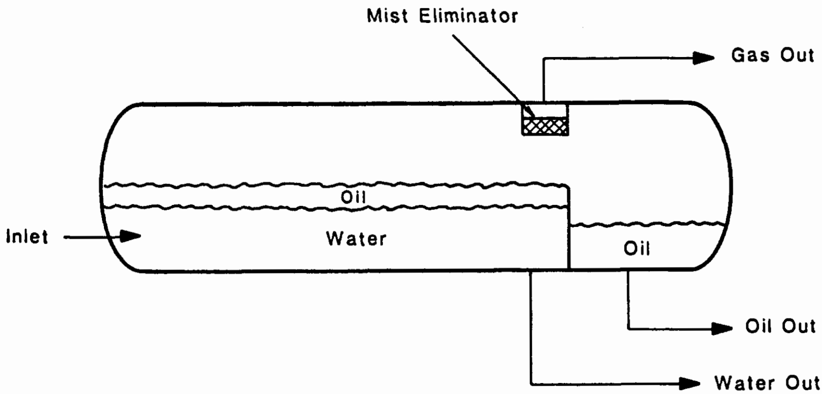

Figure 7-3. Horizontal skimmer schematic.

lesce in this section of the vessel and rise to the oil-water surface where they are captured and eventually skimmed over the oil weir. The height of the oil can be controlled by interface control, by a water leg similar to that shown in Figure 7-2, or by a bucket and weir arrangement.

Horizontal vessels are more efficient at water treating because the oil droplets do not have to flow countercurrent to the water flow. However, vertical skimmers are used in instances where:

1. Sand and other solid particles must be handled. This can be done in vertical vessels with either the water outlet or a sand drain off the bottom. Experience with elaborately designed sand drains in large horizontal vessels has not been very satisfactory.
2. Liquid surges are expected. Vertical vessels are less susceptible to high level shutdowns due to liquid surges. Internal waves due to surging in horizontal vessels can trigger a level float even though the volume of liquid between the normal operating level and the high level shutdown is equal to or larger than that in a vertical vessel. This possibility can be minimized through the installation of stilling baffles in the vessel.

The choice of pressure versus atmospheric vessel for the skimmer tank is not determined solely by the water treating requirements. The overall needs of the system need to be considered in this decision. Pressure vessels are more expensive. However, they are recommended where:

1. Potential gas blowby through the upstream vessel dump system could create too much back-pressure in an atmospheric vent system.

2. The water must be dumped to a higher level for further treating and a pump would be needed if an atmospheric vessel were installed.

Due to the potential danger from overpressure and potential gas venting problems associated with atmospheric vessels, pressure vessels are preferred. However, an individual cost/benefit decision must be made.

A minimum residence time of 10 to 30 minutes should be provided to assure that surges do not upset the system and to provide for some coalescence. As previously discussed, the potential benefits of providing much more residence time will probably not be cost efficient beyond this point. Skimmers with long residence times require baffles to attempt to distribute the flow and eliminate short circuiting. Tracer studies have shown that skimmer tanks, even those with carefully designed spreaders and baffles, exhibit poor flow behavior and short circuiting. This is probably due to density and temperature differences, deposition of solids, corrosion of spreaders, etc.

Skimmer Sizing Equations

Horizontal Cylindrical Vessel One Half Full

The required diameter and length of a horizontal cylinder operating one-half full of water can be determined from Stokes' Law as follows:

$$d\,L_{eff} = \frac{1{,}000\,Q_w \mu}{(\Delta S.G.)\,(d_m)^2} \qquad (7-3)$$

where d = vessel diameter, in.
Q_w = water flow rate, bwpd
μ = water viscosity, cp
d_m = oil droplet diameter, micron
L_{eff} = effective length in which separation occurs, ft
$\Delta S.G.$ = difference in specific gravity between the oil and water relative to water

Derivation of Equation 7-3

Oil droplets must settle vertically upward through horizontally flowing water. t_o and t_w are in s, d_m in micron, μ in cp and d in inches.

$$t_o = t_w$$

$$t_o = \frac{d}{(24)\,V_o}$$

$$V_o = \frac{1.78 \times 10^{-6} (\Delta S.G.)\, d_m^2}{\mu}$$

L_{eff} is in feet, Q in ft^3/s, A in ft^2.

$$t_w = \frac{L_{eff}}{V_w}$$

$$V_w = \frac{Q}{A}$$

Q_w is in bpd.

$$Q = \frac{Q_w\,(5.61)}{(24)(3,600)}$$

$$A = \frac{1}{2}\,\frac{\pi d^2}{(4)(144)}$$

Use efficiency factor of 1.8 for turbulence and short circuiting.

$$d\, L_{eff} = 1,000\, \frac{Q_w\, \mu}{(\Delta S.G.)\, d_m^2}$$

Any combination of L_{eff} and d that satisfies this equation will be sufficient to allow all oil particles of diameter d_m or larger to settle out of the water. However, only those combinations that also satisfy the following retention time criteria should be chosen:

$$d^2 L_{eff} = 1.4\,(t_r)_w\, Q_w \tag{7–4}$$

where $(t_r)_w$ = retention time, minutes

This equation was derived in the chapter on two-phase separators.

The choice of correct diameter and length can be obtained by plotting d versus L_{eff} for both Equation 7-3 and 7-4. Any combination that satisfies both equations is acceptable.

Horizontal Rectangular Cross Section Tank

Similarly, the required width and length of a horizontal tank of rectangular cross section can be determined from Stokes' Law using an efficiency factor of 1.9 for turbulence and short circuiting:

$$WL_{eff} = 70\frac{Q_w \mu}{(\Delta S.G.)(d_m)^2} \tag{7-5}$$

where W = width, ft
L_{eff} = effective length in which separation occurs, ft

Derivation of Equation 7-5

Oil droplets must settle virtually upward through horizontally flowing water. t_o and t_w are in s, d_m in micron, μ in cp, L_{eff}, H and W in ft, Q in ft^3/s, A in ft^2, Q_w in bpd.

$$t_o = t_w$$

$$t_o = \frac{H}{V_o}$$

$$V_o = \frac{1.78 \times 10^{-6}(\Delta S.G.)d_m^2}{\mu}$$

$$t_w = \frac{L_{eff}}{V_w}$$

$$V_w = \frac{Q}{A}$$

$$Q = \frac{Q_w(5.61)}{(24)(3{,}600)}$$

$$A = HW$$

$$W L_{eff} = 36.5\frac{Q_w \mu}{(\Delta S.G.)d_m^2}$$

Using an efficiency factor of 1.9 for turbulence and short circuiting,

$$W L_{eff} = 70\frac{Q_w \mu}{(\Delta S.G.)d_m^2}$$

Equation 7-5 is independent of height. Typically, the height of water flow is limited to less than one-half the width to assure good flow distribution. With this assumption the following equation can be derived for retention time criteria:

$$W^2 L_{eff} = 0.008\ (t_r)_w\ Q_w \quad (7\text{–}6)$$

Derivation of Equation 7-6

t_w is in s, L_{eff} in feet, V_w in ft/s, Q in ft³/sec, Q_w in bpd; H, W and L_{eff} in feet.

$$t_w = \frac{L_{eff}}{V_w}$$

$$V_w = \frac{Q}{A}$$

$$Q = \frac{Q_w\ (5.61)}{(24)\ (3{,}600)}$$

$$A = HW$$

$$t_w = 15{,}401\ \frac{HW\ L_{eff}}{Q_w}$$

H is limited to 0.5 W; $(t_r)_w$ is in minutes.

$$W^2 L_{eff} = 0.008\ (t_r)_w\ Q_w$$

The choice of W and L that satisfies both requirements can be obtained graphically. The height of water flow, H, is set equal to 0.5 W.

Vertical Cylindrical Tank

The required diameter of a vertical cylindrical tank can be determined from:

$$d^2 = 6{,}691\ F\ \frac{Q_w\ \mu}{(\Delta S.G.)\ d_m^2} \quad (7-7)$$

F is a factor that accounts for turbulence and short circuiting. For small diameter vessels (48-inch or lower) F = 1.0. For larger diameter F depends on the type of inlet and outlet spreaders, collectors, and baffles

that are provided. Tanks 10 feet in diameter should be considered to have an F = 2.0. Tanks greater than 10 feet in diameter should be discouraged due to short circuiting.

Derivation of Equation 7-7

Oil droplets must settle vertically upward through vertically downward flowing water. V_o and V_w are in ft/sec, d_m in micron, μ in cp, Q in ft^3/s, A in ft^2, Q_w in bpd, d in inches.

$$V_o = V_w$$

$$V_o = \frac{1.78 \times 10^{-6}\,(\Delta S.G.)\,d_m^2}{\mu}$$

$$V_w = \frac{Q}{A}$$

$$Q = \frac{Q_w\,(5.61)}{(24)\,(3{,}600)}$$

$$A = \frac{\pi\,d^2}{(4)\,(144)}$$

$$d^2 = 6{,}691\,F\,\frac{Q_w\,\mu}{(\Delta S.G.)\,d_m^2}$$

The height of water column in feet can be determined from retention time requirements:

$$H = 0.7\,\frac{(t_r)_w\,Q_w}{d^2} \qquad (7-8)$$

Derivation of Equation 7-8

t_w is in s, $(t_r)_w$ in minutes, H in feet, V_w in ft/s.

$$t_w = \frac{H}{V_w}$$

$$V_w = \frac{Q}{A},\ A = \frac{\pi\,d^2}{4\,(144)}$$

$$t_w = (t_r)_w\,60$$

$$H = 0.7\,\frac{(t_r)_w\,Q_w}{d^2}$$

Plate Coalescers

Plate coalescers are skim tanks or vessels that use internal plates to improve the gravity separation process. Various configurations of plate coalescers have been devised. These are commonly called parallel plate interceptors (PPI), corrugated plate interceptors (CPI), or cross-flow separators. All of these depend on gravity separation to allow the oil droplets to rise to a plate surface where coalescence and capture occur. As shown in Figure 7-4 flow is spilt between a number of parallel plates spaced a short distance apart. To facilitate capture of the oil droplet the plates are inclined to the horizontal.

Stokes' Law should apply to oil droplets as small in diameter as 1 to 10 microns. However, field experience indicates that 30 microns sets a reasonable lower limit on the droplet sizes that can be removed. Below this size small pressure fluctuations, platform vibration, etc., tend to impede the rise of the droplets to the coalescing surface.

Parallel Plate Interceptor (PPI)

The first form of a plate coalescer was the parallel plate interceptor (PPI). This involved installing a series of plates parallel to the longitudinal axis of an API separator (a horizontal rectangular cross section skim-

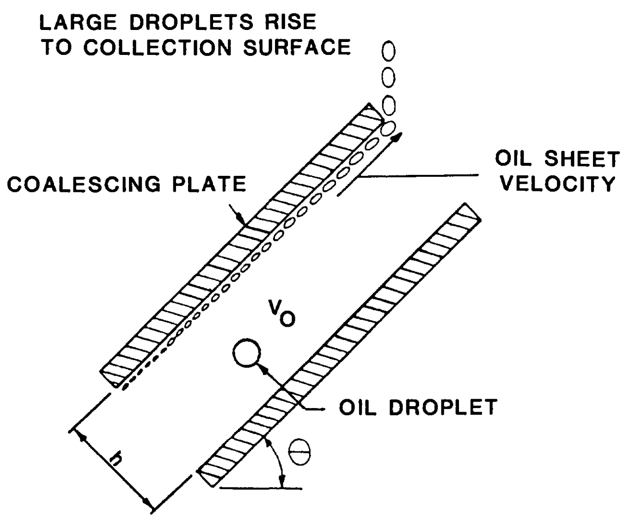

Figure 7-4. Plate coalescers.

mer) as shown in Figure 7-5. The plates form a "V" when viewed along the axis of flow so that the oil sheet migrates up the underside of the coalescing plate and to the sides. Sediments migrate towards the middle and down to the bottom of the separator, where they are removed.

Corrugated Plate Interceptor (CPI)

The most common form of parallel plate interceptor used in oil facilities is the corrugated plate interceptor (CPI). This is a refinement of the PPI in that it takes up less plan area for the same particle size removal, and has the added benefit of making sediment handling easier. Figure 7-6 illustrates a typical downflow CPI design and Figure 7-7 shows a typical CPI pack.

In CPIs the parallel plates are corrugated (like roofing material) with the axis of the corrugations parallel to the direction of flow. The plate pack is inclined at an angle of 45° and the bulk water flow is forced downward.

The oil sheet rises upward counter to the water flow and is concentrated in the top of each corrugation. When the oil reaches the end of the plate pack, it is collected in a channel and brought to the oil-water interface.

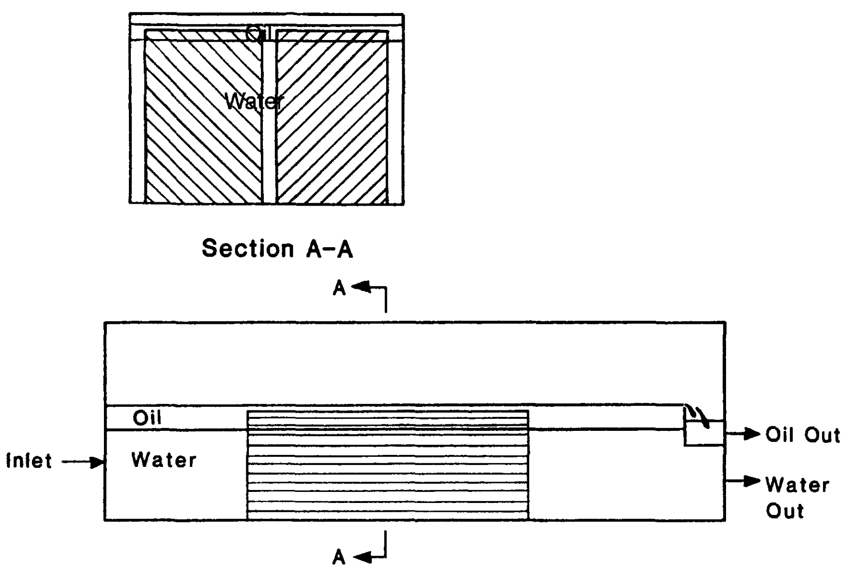

Figure 7-5. Parallel plate interceptor.

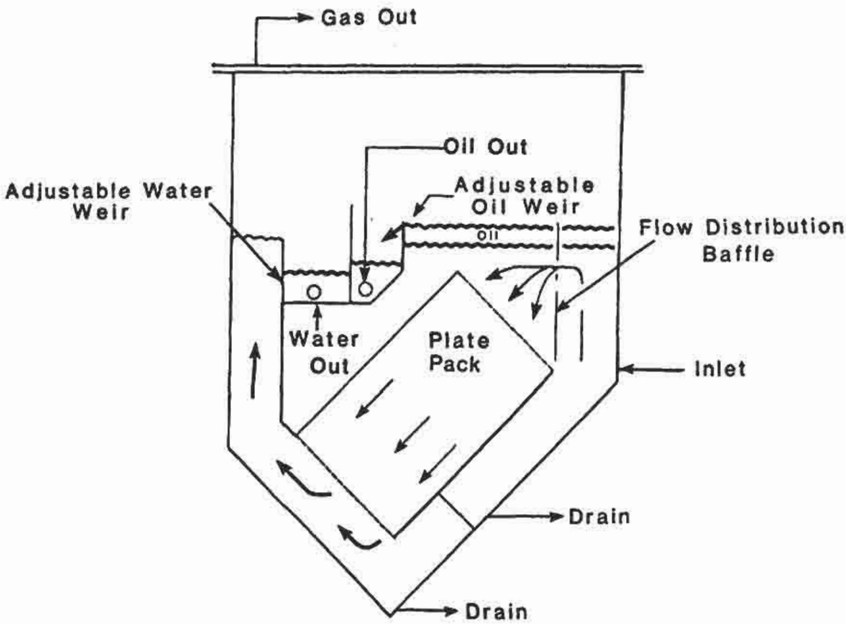

Figure 7-6. CPI flow pattern.

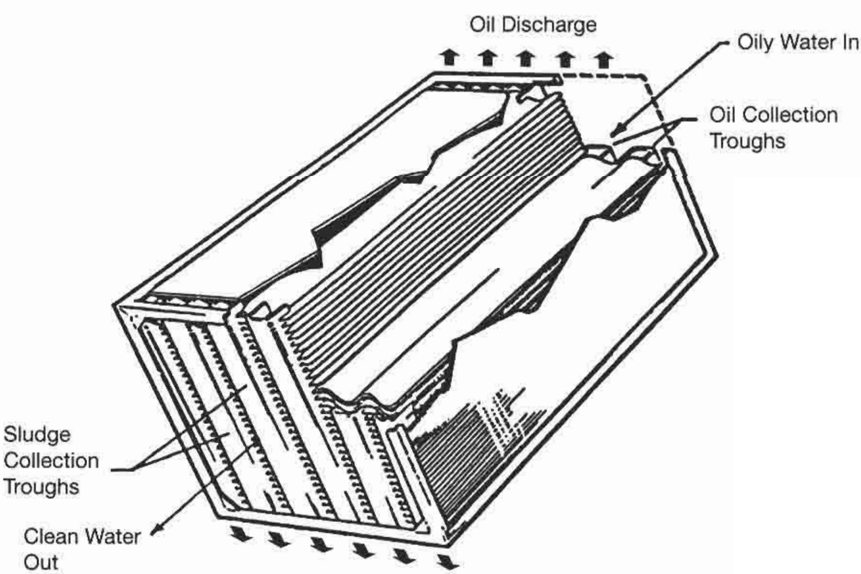

Figure 7-7. CPI plate pack.

In areas where sand or sediment production is anticipated, the sand should be removed prior to flowing through a standard CPI. Because of the required laminar flow regime all plate coalescers are efficient sand settling devices.

Experience has shown that oil wet sand may adhere to a 45° slope. Therefore, there is the possibility that the sand will adhere to and clog the plates. In addition, the sand collection channels installed at the end of the plate pack cause turbulence that affects the treating process and are themselves subject to sand plugging. To eliminate the above problems, an "upflow" CPI unit employing corrugated plates with a 60° angle of inclination may be used.

For service temperatures less than 140°F, fiberglass with a steel frame is used. For service temperatures greater than 140°F, corrosion-resistant alloys or stainless steels are recommended.

Cross-Flow Devices

Equipment manufacturers have modified the CPI configuration for horizontal water flow perpendicular to the axis of the corrugations in the plates as shown in Figure 7-8. This allows the plates to be put on a steeper angle to facilitate sediment removal, and to enable the plate pack to be more conveniently packaged in a pressure vessel. The latter benefit may be required if gas blowby through an upstream dump valve could cause relief problems with an atmospheric tank.

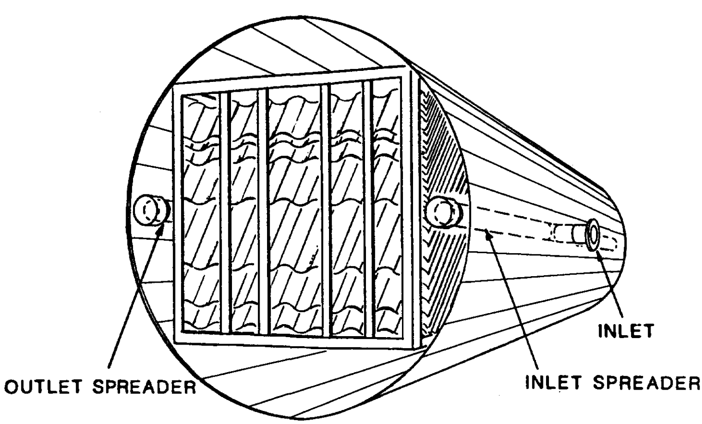

Figure 7-8. Cross-flow separator.

Cross-flow devices can be constructed in either horizontal or vertical pressure vessels. The horizontal vessels require less internal baffling as the ends of nearly each plate conduct the oil directly to the oil-water interface and the sediments to the sediment area below the water flow area. However, the pack is long and narrow and, therefore, it requires an elaborate spreader and collection device to force the water to travel across the plate pack in plug flow. There is a possibility of shearing the inlet oil droplets in the spreader, which would make separation more difficult.

Vertical units, although requiring collection channels on one end to enable the oil to rise to the oil-water interface and on the other end to allow the sand to settle to the bottom, can be designed for more efficient sand removal. CPI separators are generally cheaper and more efficient at oil removal than cross-flow separators. However, cross-flow should be considered where the use of a pressure vessel is preferred and sediment laden water is expected.

Performance

Advantages of plate coalescers are: (1) they require very little maintenance, (2) they have a smaller size and weight than skim vessels, (3) they are simple and inexpensive in comparison to some of the other types of produced-water treating devices, (4) they have no moving parts and do not require power, and (5) they are easy to install in a pressure vessel, which helps to retain hydrocarbon vapors and protect against overpressure due to failure of an upstream level control valve. Disadvantages are that plate separators are not effective for streams with slugs of oil or major surges in liquid flow rate and cannot effectively handle large amounts of solids.

Coalescer Sizing Equations

The general sizing equation for a plate coalescer with flow either parallel to or perpendicular to the scope of the plates for droplet size removal is:

$$HWL = \frac{4.8\, Q_w\, h\, \mu}{\cos\theta\, (d_m)^2\, (\Delta S.G.)} \qquad (7-9)$$

where d_m = design oil droplet diameter, micron
Q_w = bulk water flow rate, bwpd
h = perpendicular distance between plates, in.

μ = viscosity of the water, cp
θ = angle of the plate with the horizontal
H, W = height and width of the plate section perpendicular to the axis of water flow, ft
L = length of plate section parallel to the axis of water flow, ft
$\Delta S.G.$ = difference in specific gravity between the oil and water relative to water

Derivation of Equation 7-9

The oil droplet must rise to the underside of the coalescing plate. t_o and t_w in s, V_w in ft/s, Q in ft^3/s, A in ft^2, H, W and L in ft, Q_w in bpd.

$$t_o = t_w$$

$$t_w = \frac{L_{eff}}{V_w}$$

$L_{eff} = 0.7\ L$ (That is, only 70% of the actual length of the pack is effective in the settling process.)

$$V_w = \frac{Q}{A}$$

$$Q = \frac{Q_w\ (5.61)}{(24)\ (3{,}600)}$$

$A = 0.9\ H\ W$ (The plate material itself takes up 10% of the flow area.)

$$t_w = \frac{(0.7)\ L\ (0.9)\ H\ W\ (24)\ (3{,}600)}{5.61\ Q_w}$$

h is in inches.

$$t_o = \frac{h}{12 \cos\theta} \times \frac{1}{V_o}$$

$$V_o = \frac{1.78 \times 10^{-6}\ (\Delta S.G.)\ d_m^2}{\mu}$$

$$HWL = \frac{4.8\ Q_w\ h\ \mu}{\cos\theta\ (\Delta S.G.)\ d_m^2}$$

Experiments have indicated that the Reynolds number for the flow regime cannot exceed 1,600 with four times the hydraulic radius as the

characteristic dimension. Based on this correlation, the minimum H times W for a given Q_w can be determined from:

$$HW = 14 \times 10^{-4} \frac{Q_w \, h \, (S.G.)_w}{\mu} \qquad (7-10)$$

Derivation of Equation 7-10

μ' is in lb-sec/ft^2, ρ_w in lb/ft^3, R in ft, D in ft

R = hydraulic radius

$$= \frac{\text{area between plates}}{\text{wetted perimeter}}$$

$$= \frac{hW}{12} \times \frac{1}{2\,(W + h/12)} \cong \frac{h}{24}$$

Re = Reynolds number

$$Re = \frac{V_w \, D \, \rho_w}{\mu' g}$$

$$D = 4R$$

$$V_w = \frac{5.61}{(24)(3{,}600)(0.9)} \frac{Q_w}{H\,W}$$

μ is in cp

$\mu' = 2.088 \times 10^{-5} \mu$

$Re = 1{,}600$

$\rho_w = 62.4\ (S.G.)$

$$HW = 7.0 \times 10^{-4} \frac{Q_w \, h(S.G.)_w}{\mu}$$

To account for surges from control valves, use a safety factor of two. Therefore:

$$HW = 14 \times 10^{-4} \frac{Q_w \, h(S.G.)_w}{\mu}$$

CPI Sizing

For CPIs, plate packs come in standard sizes with H = 3.25 ft, W = 3.25 ft, L = 5.75 ft, h = 0.69 in., and θ = 45°. The size of the CPIs is determined by the number of standard plate packs installed. To arrive at the number of packs needed, the following equation is used:

$$\text{Number of packs} = 0.077 \frac{Q_w \mu}{(\Delta S.G.)(d_m)^2} \quad (7-11)$$

To assure that the Reynolds number limitation is met, the flow through each pack should be limited to approximately 20,000 bwpd.

It is possible to specify a 60° angle of inclination to help alleviate the solids plugging problem inherent in CPIs. This requires a 40% increase in the number of packs according to the following equation:

$$\text{Number of packs} = 0.11 \frac{Q_w \mu}{(\Delta S.G.)(d_m)^2} \quad (7-12)$$

Cross-Flow Device Sizing

Cross-flow devices obey the same general sizing equations as plate coalescers. Although some manufacturers claim greater efficiency than CPIs, the reason for this is not apparent from theory, laboratory, or field tests.

Both horizontal and vertical cross-flow separators require spreaders and collectors to uniformly distribute the water flow among the plates. For this reason, it is suggested that Equation 7-8 be modified to include a 75% spreader efficiency term so that the following equation is used:

$$HWL = \frac{6.4\, Q_w\, h\, \mu}{\cos\theta\, (\Delta S.G.)(d_m)^2} \quad (7-13)$$

Skimmer/Coalescers

Several designs that are marketed for improving oil-water separation rely on installing plates within horizontal skimmers or free-water knockouts to encourage coalescence and capture of the oil particles within the water continuous phase. The geometry of plate spacing and length can be analyzed for each of these designs using the techniques previously discussed.

Coalescing packs use a cross-flow design rather than a downflow design. The pack covers the entire inside diameter of the vessel unless sand removal internals are required. Pack lengths range from 2 to 9 feet, depending on the service. Coalescing pack materials include polypropylene, polyvinyl chloride, stainless steel, and carbon steel.

Standard spacing of cross-flow plate packs is 0.80 inches, with optional available spacing of either 0.46 or 1.33 inches. The pack is inclined

60° to lessen plugging. More coalescing sites are offered to the dispersed oil droplets due to the hexagonal pattern of the pack.

Precipitators/Coalescing Filters

In the past, it was common to direct the water to be treated through a bed of excelsior or another similar medium, as shown in Figure 7-9, to aid in the coalescing of oil droplets. However, the coalescing medium has a tendency to clog. Many of these devices in oil field service have the medium removed. In such a case they actually act like a vertical skimmer since the oil droplets must flow countercurrent to the downward flow of the water through the area where the medium was.

Coalescing filters employing sand, anthracite, or a fibrous element to catch the oil droplets and promote coalescence have been used. The filter media are designed for automatic backwash cycles. They are extremely efficient at water cleaning, but clog easily with oil and are difficult to backwash. The backwash fluid must be disposed of, which leads to further complications. Some operators have had success with sand filters in onshore operations where the backwash fluid can be routed to large settling tanks, and where the water has already been treated to 25–75 mg/l oil.

Free-Flow Turbulent Coalescers (SP Packs)

The gravity settling devices previously discussed make use of closely spaced internals to reduce the distance an oil droplet must rise to meet a coalescing surface. This is because within a tank or vessel there is very

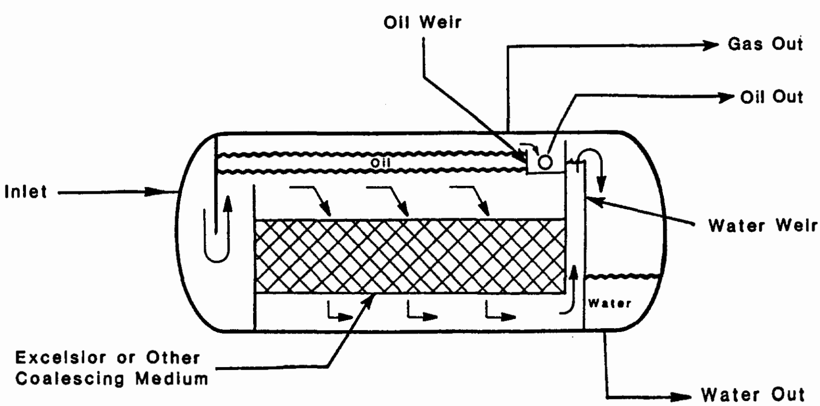

Figure 7-9. Precipitator schematic.

little turbulence to promote coalescence. The SP Pack, developed by Paragon Engineering Services and marketed by Modular Production Equipment, Houston, creates turbulent flow by forcing the water to flow through a serpentine pipe path. The path is sized to create turbulence of sufficient magnitude to cause coalescence but not so great as to shear the droplets below a specified size. The pipe path is similar in size to the inlet piping and thus is not susceptible to plugging.

As shown in Figure 7-10, the SP Pack is placed inside of any gravity settling device (skimmer, plate coalescer, etc.) and by growing a larger drop size distribution, the gravity settler is more efficient at removing oil, as shown in Figure 7-11.

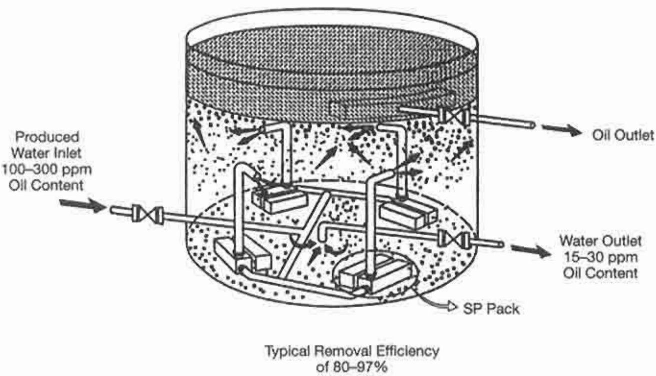

Figure 7-10. Skim tank with SP Packs installed.

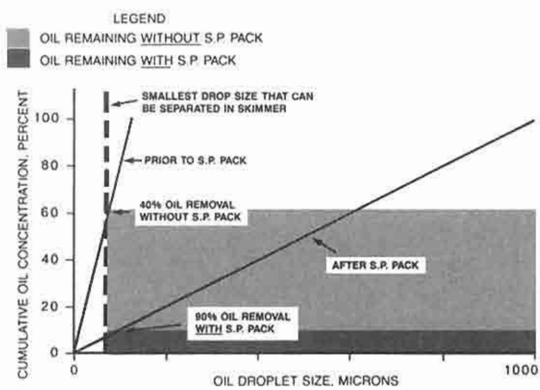

Figure 7-11. The SP pack grows a larger drop size distribution, allowing the skimmer to remove more oil.

Standard SP Pack sizes are shown in Table 7-2. They are designed to grow a drop size distribution curve with 1,000-micron maximum drop size. They are very efficient at removing oil when they are series staged, as shown in Figure 7-12. In such an installation, an enhanced drop size distribution is developed in the first pack, the larger drops are settled out. As the water flows through the next pack some of the remaining oil is coalesced, forming a second enhanced drop size distribution, and allowing more oil to be settled.

Table 7-2
Standard SP Pack Free-Flow Coalescer Sizes

Optimum[1] Flow BWPD	Model Number	Standard Inlet/Outlet Connections	Min. Tank Size	Operating[2] Range
500–1,000	LAA	1-½″	100 bbl	400–2,500
1,000–1,500	LA	2″	100 bbl	1,000–5,000
1,500–2,200	LB	3″	100 bbl	1,300–10,000
2,200–4,500	LC	4″	100 bbl	1,700–18,000
4,500–11,000	ID	6″	100 bbl	2,500–40,000
11,000–22,000	IE	8″	210 bbl	3,500–70,000
22,000–40,000	IF	10″	500 bbl	4,500–100,000
40,000–57,000	HG	12″	500 bbl	6,000–150,000

Notes:
1. *Optimum flow—range in which maximum oil removal efficiency occurs.*
2. *Operating range—broad range in which SP Pack free-flow coalescers can work without adversely affecting process performance*

All SP Pack free-flow coalescers can be fitted through STD API tank openings.

For vessels, SP Pack freeflow coalescers can be fitted through minimum 14" ID manways.

For flows in excess of 57,000 BWPD please contact MPE at Houston office.

Courtesy of Modular Production Equipment, Inc.

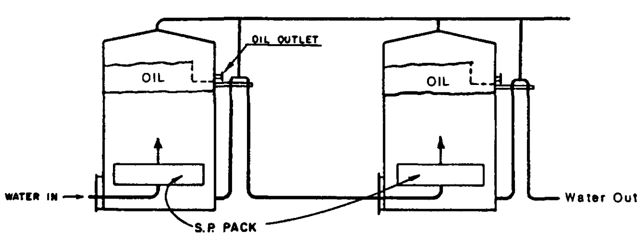

Figure 7-12. SP Packs series staged in tanks.

The efficiency in each stage is given by:

$$E = \frac{C_i - C_o}{C_i} \qquad (7-14)$$

where C_i = inlet concentration
C_o = outlet concentration

Since the drop size distribution developed by the SP Pack can be conservatively estimated as a straight line,

$$E = 1 - \frac{d_m}{d_{max}} \qquad (7-15)$$

where d_m = drop size that can be treated in the stage
d_{max} = maximum size drop created by the SP Pack. For standard SP Packs, $d_m \cong 1{,}000$ microns.

The overall efficiency of a series staged installation is then given by:

$$E_t = 1 - (1 - E)^n \qquad (7\text{–}16)$$

where n = number of stages

Flotation Units

Flotation units are the only commonly used water treating equipment that do not rely solely on gravity separation of the oil droplets. Flotation units employ a process in which fine gas bubbles are generated and dispersed in water, where they attach themselves to oil droplets or solid particulates. The gas bubbles then help to lift the oil to the water surface for collection. Flotation aids such as coagulants, polyelectrolytes, or demulsifiers are added to improve performance. Two distinct types of flotation units have been used that are distinguished by the method employed in producing the small gas bubbles needed to contact the water. These are dissolved gas units and dispersed gas units.

Dissolved Gas Units

Dissolved gas designs take a portion of the treated water effluent and saturate the water with natural gas in a contactor. The higher the pressure the more gas can be dissolved in the water. Most units are designed for a 20 to 40 psig contact pressure. Normally, 20% to 50% of the treated water is recirculated for contact with the gas. The gas saturated water is

then injected into the flotation tank as shown in Figure 7-13. The dissolved gas breaks out of solution in small diameter bubbles that contact the oil droplets in the water and bring them to the surface in a froth.

Dissolved gas units have been used successfully in refinery operations where air can be used as the gas and where large areas are available. In treating produced water for injection, it is desirable to use natural gas to exclude oxygen. This requires the venting of the gas or installation of a vapor recovery unit. Field experience with dissolved natural gas units have not been as successful as experience with dispersed gas units.

Design parameters are recommended by the individual manufacturers but normally range from 0.2 to 0.5 scf/barrel of water to be treated and flow rates of treated plus recycled water of between 2 and 4 gpm/ft^2. Retention times of 10 to 40 minutes and depths of between 6 and 9 feet are specified.

Dissolved gas units are common in chemical plant operations, but, for the following reasons, they are seldom used in producing operations: (1) they are larger than dispersed gas units and they weigh more, so they have limited application offshore, (2) many production facilities do not have vapor recovery units and, thus, the gas is not recycled, and (3) produced water has a greater tendency to cause scale in the bubble-forming device than the fresh water that is normally found in plants.

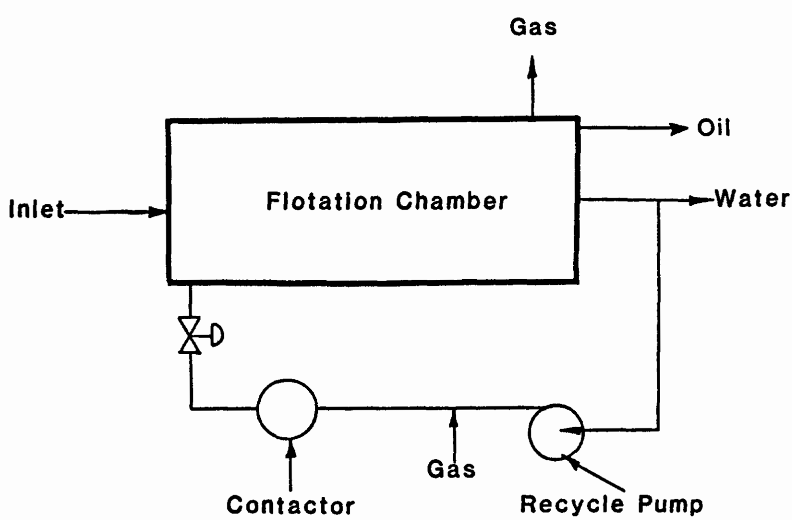

Figure 7-13. Dissolved gas flotation process.

Dispersed Gas Units

In dispersed gas units gas bubbles are dispersed in the total stream either by the use of an inductor device or by a vortex set up by mechanical rotors. Figure 7-14 shows a schematic cross section of a unit that employs a hydraulic eductor. Clean water from the effluent is pumped to a recirculation header (E) that feeds a series of venturo eductors (B). Water flowing through the eductor sucks gas from the vapor space (A) that is released at the nozzle (G) as a jet of small bubbles. The bubbles rise causing flotation in the chamber (C) forming a froth (D) that is skimmed with a mechanical device at (F).

Hydraulic eductor units are available with one, three, or four cells. These devices use less power and less gas than mechanical rotor units. Gas/water ratios are typically less than 10 ft^2/bbl at design throughput. The volume of gas dispersed in the water is not adjustable, so throughputs less than design result in higher gas/water ratios.

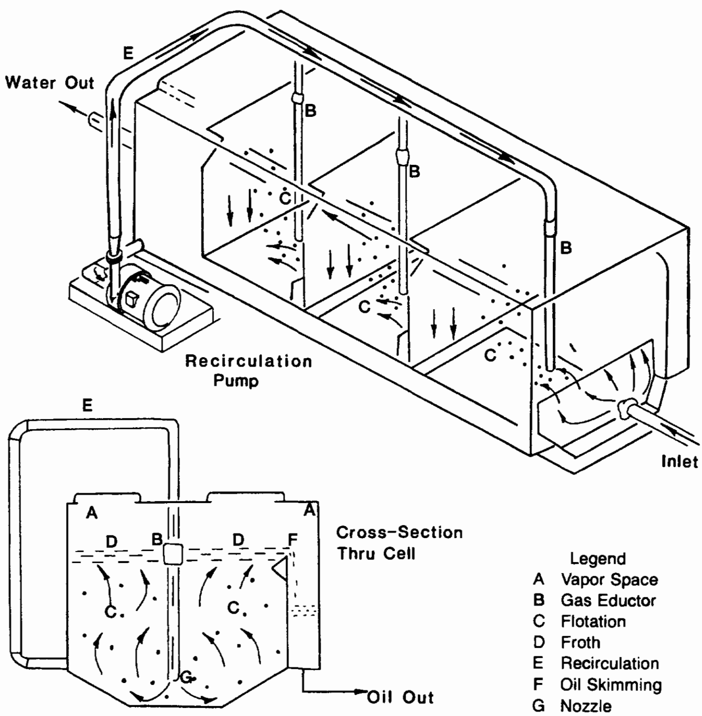

Figure 7-14. Dispersed gas flotation unit with eduction.

Figure 7-15 shows a cross section of a dispersed gas flotation cell that utilizes a mechanical rotor. The rotor creates a vortex and vacuum within the vortex tube. Shrouds assure that the gas in the vortex mixes with and is entrained in the water. The rotor and draft inducer causes the water to flow as indicated by the arrows in this plane while also creating a swirling motion. A baffle at the top directs the froth to a skimming tray as a result of this swirling motion.

Most dispersed gas units contain three or four cells. Bulk water moves in series from one cell to the other by underflow baffles. Field tests have indicated that the high intensity of mixing in each cell creates the effect of plug flow of the bulk water from one cell to the next. That is, there is virtually no short circuiting or breakthrough of a part of the inlet flow to the outlet weir box.

Sizing Dispersed Gas Units

It can be shown mathematically that an efficient design must have a high gas induction rate, a small diameter induced gas bubble, and relatively large mixing zone. The design of the nozzle or rotor, and of the internal baffles, is thus critical to the unit's efficiency.

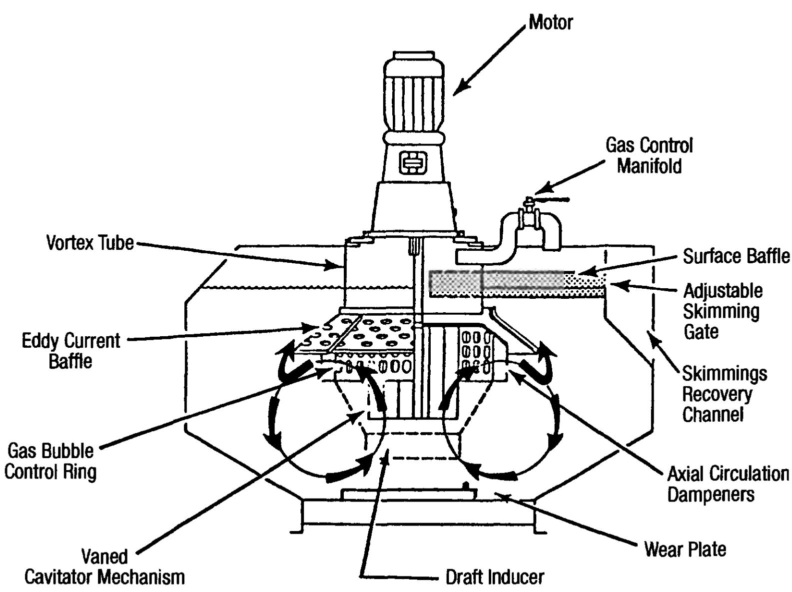

Figure 7-15. Dispersed gas flotation unit with rotor (*courtesy of Petrolite Corp.*).

As measured in actual field tests, these units operate on a constant percent removal basis. Within normal ranges their oil removal efficiency is independent of inlet concentration or oil droplet diameter.

The nozzles, rotors, and baffles for these units are patented designs. Field experiments indicate that these designs can be expected to have removal efficiencies of about 50% per cell. Each cell is designed for approximately one minute retention time to allow the gas bubbles to break free of the liquid and form the froth at the surface. Each manufacturer gives the dimensions of this standard units and the maximum flow rate based on his criteria.

From Equation 7-16, a three-cell unit can be expected to have an overall efficiency of 87% and a four-cell unit an efficiency of 94%. An efficiency of 90% is usually used for design. The unit's actual efficiency will depend on many factors that cannot be controlled or predicted in laboratory or field tests.

There are many different proprietary designs of dispersed gas units. All require a means to introduce gas into the flowstream, a mixing region where the gas can contact the oil droplets, a separation (flotation) region that allows the gas bubbles to rise to the surface, and a means to skim the froth. To operate efficiently, the unit must generate a large number of small gas bubbles. Tests indicate that bubble size decreases with increas-

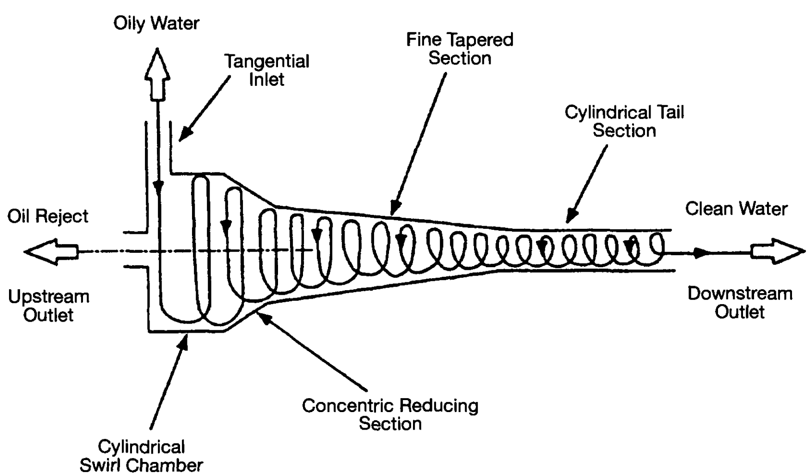

Figure 7-16. Vortoil hydrocyclone separator.

ing salinity. At salinities above 3%, bubble size appears to remain constant, but oil recovery often continues to improve.

Gas bubble/oil droplet attachment can be enhanced with the use of polyelectrolyte chemicals. These flotation aid chemicals can also be used to cause bubble/solid attachments, and thus flotation units can be used to remove solids as well.

Oil removal is dependent to some extent on oil droplet size. Flotation has very little effect on oil droplets that are smaller in diameter than 2 to 5 microns. Thus, it is important to avoid subjecting the influent to large shear forces (e.g., level control valves) immediately upstream of the unit. It is best to separate control devices from the unit by long lengths of piping (at least 300 diameters) to allow pipe coalescence to increase droplet diameter before flotation is attempted. Above 10 to 20 microns, the size of the oil droplet does not appear to affect oil recovery efficiency, and thus elaborate inlet coalescing devices are not needed.

Skimmed oily water volumes are typically 2 to 5% of the machine's rated capacity and can be as high as 10% when there is a surge of water flow into the unit. Since skimmed fluid volume is a function of weir length exposure over time, operation of the unit at less than design capacity increases the water residence time but does not decrease the skimmed fluid volumes.

Hydrocyclones

Hydrocyclones, sometimes called enhanced gravity separators, use centrifugal force to remove oil droplets from oily water. As shown in Figure 7-16, static hydrocyclones consist of the following four sections: a cylindrical swirl chamber, a concentric reducing section, a fine tapered section, and a cylindrical tail section. Oily water enters the cylindrical swirl chamber through a tangential inlet, creating a high-velocity vortex with a reverse-flowing central core. The fluid accelerates as it flows through the concentric reducing section and the fine tapered section. The fluid then continues at a constant rate through the cylindrical tail section. Larger oil droplets are separated out from the fluid in the fine tapered section, while smaller droplets are removed in the tail section. Centripetal forces cause the lighter-density droplets to move toward the low-pressure central core, where axial reverse flow occurs. The oil is removed through a small-diameter reject port located in the head of the hydrocyclone. Clean water is removed through the downstream outlet.

Static hydrocyclones require a minimum pressure of 100 psi to produce the required velocities. Manufacturers make designs that operate at lower pressures, but these models have not always been as efficient as those that operate at higher inlet pressures. If a minimum separator pressure of 100 psi is not available, a low-shear pump should be used (e.g., a progressive cavity pump) or sufficient pipe should be used between the pump and the hydrocyclone to allow pipe coalescence of the oil droplets. As is the case with flotation units, hydrocyclones do not appear to work well with oil droplets less than 10 to 20 microns in diameter.

Performance is chiefly influenced by reject ratio and pressure drop ratio (PDR). The reject ratio refers to the ratio of the reject fluid rate to the total inlet fluid rate. Typically, the optimum ratio is between 1 and 3%. This ratio is also proportional to the PDR. Operation below the optimum reject ratio will result in low oil removal efficiencies. Operation above the optimum reject ratio does not impair oil removal efficiency, but it increases the amount of liquid that must be recirculated through the facility. The PDR refers to the ratio of the pressure difference between the inlet and reject outlets and the difference from the inlet to the water outlet. A PDR of between 1.4 and 2.0 is usually desired. Performance is also affected by inlet oil droplet size, concentration of inlet oil, differential specific gravity, and inlet temperature. Temperatures greater than 80°F result in better operation.

Although the performance of hydrocyclones varies from facility to facility (as with flotation units), an assumption of 90% oil removal is a reasonable number for design. Often the unit will perform better than this, but for design it would be unwise to assume this will happen. Performance cannot be predicted more accurately from laboratory or field testing because it is dependent on the actual shearing and coalescing that occurs under field flow conditions and on impurities in the water, such as residual treating and corrosion chemicals and sand, scale and corrosion products, which vary with time.

Hydrocyclones are excellent coalescing devices, and they actually function best as a primary treating device followed by a downstream skim vessel that can separate the 500 to 1,000 micron droplets that leave with the water effluent. A simplified P&ID for a hydrocyclone is shown in Figure 7-17.

Advantages of static hydrocyclones include: (1) they have no moving parts (thus, minimum maintenance and operator attention is required), (2)

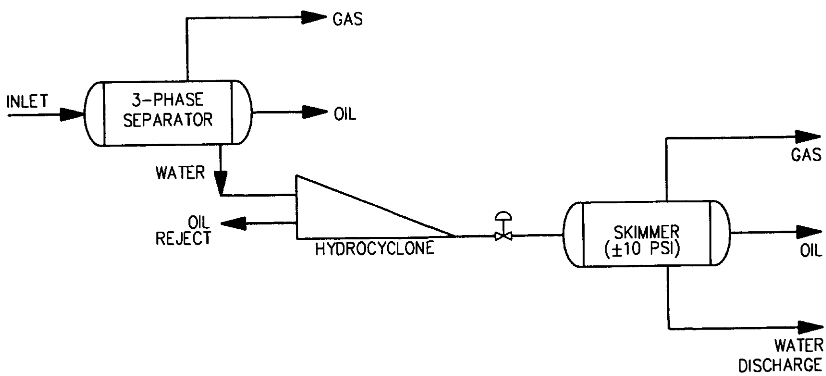

Figure 7-17. P&ID for a typical hydrocyclone system.

their compact design reduces weight and space requirements when compared to those of a flotation unit, (3) they are insensitive to motion (thus, they are suitable for floating facilities), (4) their modular design allows easy addition of capacity, and (5) they offer lower operating costs when compared to flotation units, if inlet pressure is available.

Disadvantages include the need to install a pump if oil is available only at low pressure and the tendency of the reject port to plug with sand or scale. Sand in the produced water will cause erosion of the cones and increase operating costs.

Disposal Piles

Disposal piles are large diameter (24- to 48-inch) open-ended pipes attached to the platform and extending below the surface of the water. Their main uses are to (1) concentrate all platform discharges into one location, (2) provide a conduit protected from wave action so that discharges can be placed deep enough to prevent sheens from occurring during upset conditions, and (3) provide an alarm or shutdown point in the event of a failure causing oil to flow overboard.

Most authorities having jurisdiction require all produced water to be treated (skimmer tank, coalescer, or flotation) prior to disposal in a disposal pile. In some locations, disposal piles are permitted to collect treated produced water, treated sand, liquids from drip pans and deck drains, and as a final trap for hydrocarbon liquids in the event of equipment upsets.

Disposal piles are particularly useful for deck drainage disposal. This flow, which originates either from rainwater or washdown water, typically contains oil droplets dispersed in an oxygen-laden fresh or saltwater phase. The oxygen in the water makes it highly corrosive and commingling with produced water may lead to scale deposition and plugging in skimmer tanks, plate coalescers, or flotation units. The flow is highly irregular and would thus cause upsets throughout these devices. Finally, this flow must gravitate to a low point for collection and either be pumped up to a higher level for treatment or treated at that low point. Disposal piles are excellent for this purpose. They can be protected from corrosion, they are by design located low enough on the platform to eliminate the need for pumping the water, they are not severely affected by large instantaneous flow rate changes (effluent quality may be affected to some extent but the operation of the pile can continue), they contain no small passages subject to plugging by scale buildup, and they minimize commingling with the process since they are the last piece of treating equipment before disposal.

Disposal Pile Sizing

The produced water being disposed of has been treated in vessels having the capability of treating smaller droplets than those that can be predicted to settle out in the relatively slender disposal pile. Small amounts of separation will occur in the disposal pile due to coalescence in the inlet piping and in the pipe itself. However, no significant treating of produced water can be expected.

Most authorities having jurisdiction require that deck drainage be disposed of with no free oil. If the deck drainage is merely contaminated rain water, the disposal pile diameter can be estimated from the following equation, assuming the need to separate 150-micron droplets:

$$d^2 = \frac{0.3\,(Q_w + 0.356\,A_D R_w + Q_{WD})}{(\Delta S.G.)} \tag{7-17}$$

where d = pile internal diameter, in.

Q_w = produced water rate (if in disposal pile), bwpd

A_D = plan area of the deck, ft^2

R_w = rainfall rate, in./hr

= 2 in./hr for Gulf of Mexico

$\Delta S.G.$ = difference in specific gravity between oil droplets and rain water

Q_{WD} = washdown rate, bpd
= 1,500 N
N = number of 50 gpm washdown hoses

Derivation of Equation 7-17

From the equation for a vertical skim tank with F = 1.0:

$$d^2 = 6,691 \frac{Q_w \mu_w}{(\Delta S.G.)\, d_m^2}$$

d_m = 150 microns

Q_w = produced water rate if it is disposed in pile + rainfall rate or washdown rate, bpd

$$Q_R = \frac{R_w}{12} A_D \frac{(24)}{(5.61)}$$

Q_R = rainfall rate, bpd

R_w = rainfall rate, in./hr

A_D = deck area, ft^2

$$d^2 = \frac{0.3\,(Q_w + 0.356\, A_D R_w + Q_{WD})}{(\Delta S.G.)}$$

In Equation 7-17 either the washdown rate or the rainfall rate should be used as it is highly unlikely that both would occur at the same time. The produced water rate is only used if produced water is routed to the pile for disposal.

The disposal pile length should be as long as the water depth permits in shallow water to provide for maximum oil containment in the event of a malfunction and to minimize the potential appearance of any sheen. In deep water, the length is set to assure that an alarm and then a shutdown signal can be measured before the pile fills with oil.

These signals must be high enough so as not to register tide changes. The length of pile submergence below the normal water level required to assure that a high level will be sensed before the oil comes within 10 feet of the bottom is given by:

$$L = \frac{(H_T + H_S + H_A + H_{SD})\, S.G._o}{(\Delta S.G.)} + 10 \qquad (7-18)$$

where L = depth of pile below MWL (submerged length), ft
H_t = normal tide range, ft
H_s = design annual storm surge, ft
H_A = alarm level (usually 2 ft), ft
H_{SD} = shutdown level (usually 2 ft), ft
$(S.G.)_o$ = specific gravity of the oil relative to water

Derivation of Equation 7-18

$$(H_w + H_T + H_S + H_A + H_{SD})\rho_o = H_w\rho_w$$

$$(H_w)S.G._o + (H_T + H_S + H_A + H_{SD})S.G._o = (H_w)S.G._w$$

$$H_W = \frac{(H_T + H_S + H_A + H_{SD})\,S.G._o}{(\Delta S.G.)}$$

For pile length 10 feet longer than oil column height.

$$L = H_W + 10 = \frac{(H_T + H_S + H_A + H_{SD})\,S.G._o}{(\Delta S.G.)} + 10$$

It is possible in shallow water to measure the oil-water interface for alarm or shutdown with a bubble arrangement and a shorter pile. However, this is not recommended where water depth permits a longer pile. To minimize wave action effects a minimum pile length of about 50 feet is required.

Skim Pile

The skim pile is a type of disposal pile. As shown in Figure 7-18, flow through the multiple series of baffle plates creates zones of no flow that reduce the distance a given oil droplet must rise to be separated from the main flow. Once in this zone, there is plenty of time for coalescence and gravity separation. The larger droplets then migrate up the underside of the baffle to an oil collection system.

Besides being more efficient than standard disposal piles, from an oil separation standpoint, skim piles have the added benefit of providing for some degree of sand cleaning. Most authorities having jurisdiction state that produced sand must be disposed of without "free oil." It is doubtful that sand from a vessel drain meets this criterion when disposed of in a standard disposal pile.

Sand traversing the length of a skim pile will abrade on the baffles and be water washed. This can be said to remove the free oil that is then captured in a quiescent zone.

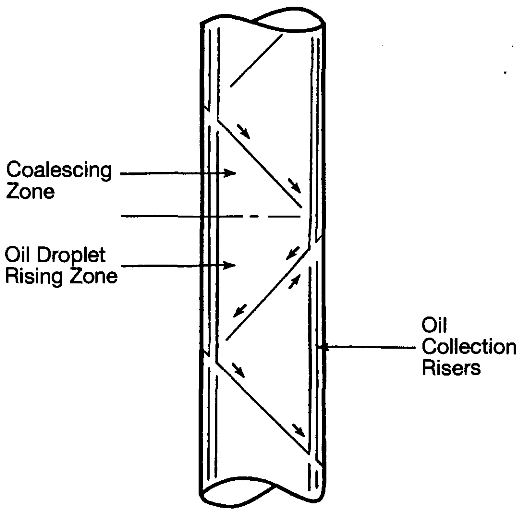

Figure 7-18. Skim pile flow pattern (*courtesy of Engineering Specialists Inc.*).

Skim Pile Sizing

The determination of skim pile length is the same as that for any other disposal pile. Because of the complex flow regime a suitable equation has yet to be developed to size skim piles for deck drainage. However, field experience has indicated that acceptable effluent is obtained with 20 minutes retention time in the baffled section of the pile. Using this and assuming that 25% of the volume is taken up by the coalescing zones:

$$d^2L' = 19.1(Q_w + 0.356\, A_D R_w + Q_{WD}) \qquad (7\text{–}19)$$

where L' = Length of baffle section, ft. The submerged length is $L' + 15$ feet to allow for an inlet and exit from the baffle section.

Derivation of Equation 7-19

t_w is in s, Q in ft^3/s, d in inches, Vol in ft^3, Q_w in bpd, L' in feet.

$$t_w = \frac{\text{Vol}}{Q}$$

$$\text{Vol} = \frac{\pi\, d^2 L'}{(4)(144)} \times \frac{3}{4}$$

$$Q = \frac{5.61}{(24)(3{,}600)}(Q_W + 0.356\, A_D R_W + Q_{WD})$$

$t_w = 60(t_r)_w = 1{,}200$

$d^2L' = 19.1(Q_w + 0.356\ A_D R_w + Q_{WD})$

DRAIN SYSTEMS

A drain system that is connected directly to pressure vessels is called a "pressure" or "closed" drain system. A drain system that collects liquids that spill on the ground is an "atmospheric," "gravity," or "open" drain. The liquid in a closed drain system must be assumed to contain dissolved gases that flash in the drain system and can become a hazard if not handled properly. In addition, it must be assumed that a closed drain valve could be left open by accident. Once the liquid has drained out of the vessel, a large amount of gas will flow out of the vessel into the closed drain system (gas blowby) and will have to be handled safely. Thus, closed drain systems should always be routed to a pressure vessel and should never be connected to an open drain system.

Liquid gathered in an open drain system is typically rain water or washdown water contaminated with oil. With the oil usually circulated back into the process, every attempt should be made to minimize the amount of aerated water that is recycled with the oil. This goal is best achieved by routing open drains to a sump tank that has a gas blanket and operates as a skimmer. To keep gas from the skimmer from flowing out the drain, a water seal should be built into the inlet to the sump tank. Water seals should also be installed on laterals from separate buildings or enclosures to keep the open drain system from being a conduit of gas from one location in the facility to another.

INFORMATION REQUIRED FOR DESIGN

1. *Effluent quality.* The Environmental Protection Agency (EPA) establishes the maximum amount of oil and grease content in water that can be discharged into navigable waters of the United States. In other locations local governments or governing bodies will establish this criterion.

   Examples of worldwide produced water effluent oil concentration limitations include:

Ecuador, Colombia, Brazil, Argentina, Venezuela:	30 mg/l all facilities

Indonesia:	15 mg/l new facilities
	25 mg/l "grandfathered" facilities
Malaysia, Middle East:	30 mg/l all facilities
Nigeria, Angola, Cameroon, Ivory Coast:	50 mg/l all facilities
North Sea, Australia:	30 mg/l all facilities
Thailand:	50 mg/l all facilities
USA:	29 mg/l OCS waters
	Zero discharge inland waters

2. *Produced water flow rate* (Q_w, bwpd).
3. *Specific water gravity of produced water* $(S.G.)_w$. Assume 1.07 if data are not available.
4. *Waste water viscosity at flowing temperatures* (μ, cp). Assume 1.0 cp if data are not available.
5. *Concentration of oil in water to be treated* (mg/l or ppm). This is best determined from field samples or laboratory data.
6. *Specific gravity of oil at flowing temperature* $(S.G.)_o$.
7. *Particle size distribution curve for oil droplets in the produced water.* This is best determined from field samples or laboratory data and an analysis of drop size management due to dispersion and coalescence in the piping system.
8. *Design rainfall rate* (R_w, in./hr). Assume 2 in./hr in the Gulf of Mexico.
9. *Flow rate for washdown* (Q_{WD}, bpd). Assume 1,500 bwpd per 50 gpm washdown hose.
10. *Particle size distribution curve for "free oil" droplets in deck drainage.*
11. *Concentration of soluble oil at discharge conditions* (mg/l or ppm).

INFLUENT WATER QUALITY

Produced Water

The first step in choosing a water treating system is to characterize the influent water streams. It is necessary to know both the *oil concentration* in this stream and the *particle size distribution* associated with this concentration. This is best determined from field samples and laboratory data.

Various attempts have been made to develop design procedures to determine oil concentration in water outlets from properly designed free-water knockouts and treaters. A conservative assumption would be that the water contains less than 1,000 to 2,000 mg/l of dispersed oil.

It is possible to theoretically trace the particle size distribution up the tubing, through the choke, flowlines, manifolds and production equipment into the free-water knockout using equations presented in previous sections. However, many of the parameters needed to solve these equations, especially those involving coalescence, are unknown.

Because of the dispersion through the water dump valve, the oil size distribution at the outlet of a free-water knockout or heater treater is not a significant design parameter. From the dispersion theory it can be shown that after passing through the dump valve a maximum droplet diameter on the order of 10 to 50 microns will exist no matter what the droplet size distribution was upstream of this valve.

If there were sufficient time for coalescence to occur in the piping downstream of the dump valve, then the maximum droplet diameter would be defined by Equation 7-2 prior to the water entering the first vessel in the water treating system.

The solution of this equation requires the determination of surface tension. The surface tension of an oil droplet in a water continuous phase is normally between 1 and 50 dynes/cm. It is not possible to predict the value without actual laboratory measurements in the produced water. Small amounts of impurities in the produced water can lower the surface tension significantly from what might be measured in synthetic water. In addition, as these impurities change with time, so will the surface tension. In the absence of data it is recommended that a maximum diameter of between 250 and 500 microns be used for design.

It is clear that there will be distribution of droplet sizes from zero to the maximum size, and this distribution will depend upon parameters unknown at the time of initial design. Experimental data indicate that a conservative assumption for design would be to characterize the distribution by a straight line as shown in Figure 7-19.

Soluble Oil

In every system substances that show up as "oil" in the laboratory test procedure will be dissolved in the water. This is especially true where samples are acidized for "stabilization" prior to extraction with a sol-

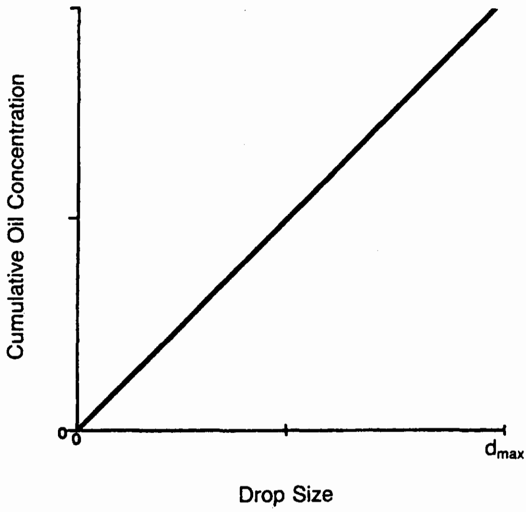

Figure 7-19. Droplet size distribution for design.

vent. This soluble oil cannot be removed by the systems discussed in this chapter. The soluble oil concentration should be subtracted from the discharge criteria to obtain a concentration of dispersed oil for design. Soluble oil concentrations as high as 1,000 mg/l have been measured on rare occasions.

Deck Drainage

Federal regulations and most authorities having jurisdiction require that "free oil" be removed from deck drainage prior to disposal. It is extremely difficult to predict an oil drop size distribution for rainwater or washdown water that is collected in an open drain system, and regulations do not define what size droplet is meant by "free oil."

Long-standing refinery practice is to size the drain water treating equipment to remove all oil droplets 150 microns in diameter or larger. If no other data are available, it is recommended that this be used in sizing sumps and disposal piles.

EQUIPMENT SELECTION PROCEDURE

It is desirable to bring information included earlier into a format that can be used by the design engineer in selecting and sizing the individual

pieces of equipment needed for a total water treating system. Federal regulations and most authorities having jurisdiction require that produced water from the free-water knockout receive at least some form of primary treatment before being sent to a disposal pile or skim pile. Deck drainage may be routed to a properly sized disposal pile that will remove "free oil."

Every water treating system design must being with the sizing, for liquid separation of a free-water knockout, heater treater, or three-phase separator. These vessels should be sized in accordance with the procedures discussed in previous chapters.

With the exception of these restraints the design engineer is free to arrange the system as he sees fit. There are many potential combinations of the equipment previously described. Under a certain set of circumstances, it may be appropriate to dump the water from a free-water knockout directly to a skim tank for final treatment before discharge. Under other circumstances, a full system of plate coalescers, flotation units, and skim piles may be needed. In the final analysis the choice of a particular combination of equipment and their sizing must rely rather heavily on the judgment and experience of the design engineer. The following procedure is meant only as a guideline and not as a substitute for this judgment and experience. Many of the correlations presented herein should be refined as new data and operating experience become available. In no instance is this procedure meant to be used without proper weight given to operational experience in the specific area.

1. Determine the oil content of the produced water influent. In the absence of other information 1,000 to 2,000 mg/l could be assumed.
2. Determine dispersed oil effluent quality. In the absence of other information, use 23 mg/l for design in the Gulf of Mexico and other similar areas (29 mg/l allowed less 6 mg/l dissolved oil).
3. Determine oil drop size distribution in the influent produced water stream. Use a straight-line distribution with a maximum diameter of 250 to 500 microns in the absence of better data.
4. Determine oil particle diameter that must be treated to meet effluent quality required. This can be calculated as effluent quality divided by influent quality times the maximum oil particle diameter calculated in step 3.
5. If there is a large amount of space available (as in an onshore location), consider an SP Pack system. Proceed to step 10. If the answer

to step 4 is less than 30 to 50 microns, a flotation unit or cyclone is needed. Proceed to step 6. If the answer to step 4 is greater than 30 microns, a skim tank or plate coalescer could be used as a single stage of treatment, but this is not really recommended. Proceed to step 9.

6. Determine flotation cell influent quality from the required effluent quality assuming 90% removal. Influent quality is effluent quality desired times 10.
7. If required flotation cell influent quality is less than quality determined in step 1, determine the particle diameter that must be treated in skim tank or plate coalescer to meet this quality. This can be calculated as the flotation cell influent quality divided by the influent quality determined in step 1 times the maximum particle diameter calculated in step 3.
8. Determine effluent from hydrocyclone, assuming that it is 90% efficient, and determine the particle diameter that must be treated in the downstream skim vessel, assuming that $d_{max} = 500$. This value can be calculated as 500 times the dispersed oil effluent quality (step 1) divided by the effluent concentration from the hydrocyclone.
9. Determine skimmer dimensions.
   a. Choose horizontal or vertical configuration.
   b. Choose pressure vessel or atmospheric vessel.
   c. Determine size. Refer to appropriate equations.
10. Determine overall efficiency required, efficiency per stage and number of stages for an SP Pack system.
11. Determine plate coalescer dimensions.
    a. Choose CPI or cross-flow configuration.
    b. Determine size. Refer to appropriate equations.
12. Choose skim tank, SP Pack, or plate coalescer for application, considering cost and space available.
13. Choose method of handling deck drainage.
    a. Determine whether rainwater rate or washdown rate governs design.
    b. Size disposal pile for drainage assuming 150-micron drop removal. Refer to appropriate equations.

c. If disposal pile diameter too large,
   i. Size sump tank to use with disposal pile (refer to appropriate equations), or
   ii. Size skim pile (refer to appropriate equations).

EQUIPMENT SPECIFICATION

Once the equipment types are selected using the previous procedure, the design equations presented in Chapter 6 can be used to specify the main size parameters for each of the equipment types.

Skim Tank

1. Horizontal vessel design—The internal diameter and seam-to-seam length of the vessel can be determined. The effective length of the vessel can be assumed to be 75% of the seam-to-seam length.
2. Vertical vessel design—The internal diameter and height of the water column can be determined. The vessel height can be determined by adding approximately 3 feet to the water column height.

SP Pack System

The number and size of tanks can be determined. Alternatively, the dimensions and number of compartments in a horizontal flume can be specified.

CPI Separator

The number of plate packs can be determined.

Cross-Flow Devices

The acceptable dimensions of plate pack area can be determined. The actual dimensions depend on the manufacturers' standard sizes.

Flotation Cell

Information is given to select a size from the manufacturers' data.

Disposal Pile

The internal diameter and length can be determined. For a skim pile the length of the baffle section can be chosen.

Example 7-1: Design the Produced Water Treating System for the Data Given.

Given: 40° API
5,000 bwpd
Deck size is 2,500 ft^2
48 mg/l discharge criteria
Water gravity-feeds to system

Step 1. Assume 6 mg/l soluble oil, and oil concentration in produced water is 1,000 mg/l.

Step 2. Effluent quality required is 48 mg/l. Assume 6 mg/l dissolved oil. Therefore, effluent quality required is 42 mg/l.

Step 3. Assume maximum diameter of oil particle (d_{max}) = 500 microns.

Step 4. Using Figure 7-19, the size of oil droplet that must be removed to reduce the oil concentration from 1,000 mg/l to 42 mg/l is:

$$\frac{d_m}{500} = \frac{42}{1,000}$$

$$d_m = 21 \text{ microns}$$

Step 5. Consider an SP series tank treating system. See step 10. If SP packs are not used, since d_m is less than 30 microns a flotation unit or hydrocyclone must be used. Proceed to step 6. (Note: since d_m is close to 30 microns, it may be possible to treat this water without a flotation unit. We will take the more conservative case for this example.)

Step 6. Since the flotation cell is 90% efficient in order to meet the design requirements of 42 mg/l it will be necessary to have an influent quality of 420 mg/l. This is lower than the 1,000 mg/l concentration in the produced water assumed to in step 1. Therefore, it is necessary to install a primary treating device upstream of the flotation unit.

Step 7. Using Figure 7-19, the size of oil droplet that must be removed to reduce the oil concentration from 1,000 mg/l to 420 mg/l is:

$$\frac{d_m}{500} = \frac{420}{1{,}000}$$

$$d_m = 210 \text{ microns}$$

Step 8. Inlet to water treating system is at too low a pressure for a hydrocyclone. Size a skim vessel upstream of the flotation unit.

Step 9. Skim vessel design. Pressure vessel needed for process considerations (e.g., fluid flow, gas blowby).

a. Assume horizontal pressure vessel.

Settling equation

$$d\,L_{eff} = \frac{1{,}000\,Q_w \mu_w}{(\Delta S.G.)\,(d_m)^2}$$

$$\mu_w = 1.0 \text{ (assumed)}$$

$$(S.G.)_w = 1.07 \text{ (assumed)}$$

$$(S.G.)_o = 0.83 \text{ (calculated)}$$

$$d\,L_{eff} = \frac{(1{,}000)(5{,}000)(1.0)}{(0.24)(210)^2} = 472$$

Assume various diameters (d) and solve for L_{eff}.

d in.	L_{eff} ft	Actual Length ft
24	19.7	26.3
48	9.8	13.1
60	7.9	10.5

Retention time equation

Assume retention time of 10 minutes.

$$d^2 L_{eff} = 1.4\,(t_r)_w Q_w$$

$$d^2 L_{eff} = (1.4)(10)(5{,}000)$$

d in.	L_{eff} ft	Actual Length ft
48	30.4	40.4
72	13.5	17.9
84	9.9	13.1
96	7.6	10.1

b. Assume vertical pressure vessel

Settling equation

$$d^2 = 6{,}691\,F\frac{Q_w\mu_w}{(\Delta S.G.)\,(d_m)^2}$$

F = 1.0 assumed

$$d^2 = \frac{(6{,}691)\,(1.0)\,(5{,}000)\,(1.0)}{(0.24)\,(210)^2}$$

d = 56.22 in.

Retention time equation

$$H = 0.7\,\frac{(t_r)_w Q_w}{d^2} \approx 0.7\,\frac{(10)\,(5{,}000)}{d^2}$$

d in.	H ft	Seam-to-Seam Height ft
60	9.72	12.7
66	8.03	11.0
72	6.75	9.8

A vertical vessel 60-in. × 12.5-ft or 72-in. × 10-ft would satisfy all the parameters. Depending on cost and space considerations, recommend 72-in. × 10-ft vertical skimmer vessel for this application.

Step 10. Investigate SP Packs in tanks as an option. Calculate overall efficiency required:

$$E_t = \frac{1{,}000 - 42}{1{,}000} = 0.958$$

Assume 10-ft-diameter vertical tanks:

$$d_m^2 = 6,691\, F \frac{Q_w\, \mu_w}{(\Delta S.G.)\,(d^2)}$$

$$d_m^2 = \frac{6,691\,(2)\,(5,000)\,(1.0)}{(0.24)\,(120)^2}$$

$$d_m = 139$$

Assume SP Pack grows 1,000 micron drops:

$$E = 1 - \frac{d_m}{1,000}$$

$$E = 0.861$$

One acceptable choice is two 10-ft-diameter SP tanks in series.

$$E_t = 1 - (1 - 0.861)^2 = 0.981$$

Step 11. Check for alternate selection of CPI.

$$\text{Number of packs} = 0.077 \frac{Q_w \mu_w}{(\Delta S.G.)\, d_m^2}$$

$$= \frac{(0.077)\,(5,000)\,(1.0)}{(0.24)\,(210)^2}$$

$$= 0.04 \text{ packs}$$

$Q_w < 20,000$: use 1 pack CPI

Step 12. Recommended skimmer vessel over CPI as skimmer will take up about same space, cost less, and not be susceptible to plugging. Note that it would also be possible to investigate other configurations such as skim vessel, SP Pack, CPI, etc., as alternatives to the use of a flotation unit.

Step 13. Sump design: Sump is to be designed to handle the maximum of either rain water or washdown hose rate.

a. Rain water rate:

Assume R_w = Rainfall rate; 2 in./hr.
A_D = Deck area; 2500 ft^2
$Q_w = 0.356\, A_D\, R_W$

$$Q_w = (0.356)(2{,}500)(2)$$
$$= 1{,}780 \text{ BWPD}$$

b. Washdown rate:

Assume $N = 2$

$$Q_{WD} = 1{,}500\ N$$
$$Q_{WD} = 1{,}500\ (2)$$
$$Q_{WD} = 3{,}000 \text{ BWPD}$$

The minimum design usually calls for two hoses.

Because fresh water enters the sump via the drains, the sump tank must be sized using a specific gravity of 1.0 and a viscosity of 1.0 for fresh water.

c. Assume horizontal rectangular cross-section sump.

Settling equation:

$$WL_{eff} = 70 \frac{Q_w\, \mu_w}{(\Delta S.G.)\, d_m^2}$$

$$WL_{eff} = 70 \frac{(3{,}000)\,(1.0)}{(0.150)\,(150)}$$

$$WL_{eff} = 62.2$$

W = Width, ft
L_{eff} = Effective length in which separation occurs, ft
H = Height of tank, which is 1.5 times higher than water level within tank, or 0.75 W

Tank Width ft	Tank L_{eff} ft	Seam-to-Seam Length (1.2 L_{eff})	Height ft
4	15.6	20.2	3.0
5	12.4	16.2	3.8
6	10.4	13.5	4.5

A horizontal tank 6 ft × 14 ft × 5 ft would satisfy all design parameters.

d. If it is determined that the dimensions of the sump tank are inappropriately large for the platform, an SP Pack can be added upstream of the sump tank to increase oil droplet size by approximately two times the inlet droplet size.

Therefore, the sump tank size with an SP Pack can be determined by:

$$WL_{eff} = \frac{70\,(3{,}000)\,(1.0)}{(0.15)\,(300)^2}$$

$$WL_{eff} = 15.6$$

W = Width, ft
L_{eff} = Effective length in which separation occurs, ft
H = Height of tank, which is 1.5 times higher than water level within tank, or 0.75W

Tank Width ft	**Tank L_{eff} ft**	**Seam-to-Seam Length (1.2 L_{eff})**	**Height ft**
3	5.2	6.7	2.3
4	3.9	5.1	3.0
5	3.1	4.0	3.8

A horizontal tank (with an SP Pack) 4 ft × 4 ft × 5 ft would satisfy all design parameters. It can be seen that by adding an SP Pack, sump tank sizes can be substantially reduced.

Step 14. Recovered Oil Tank:

Assume a cylindrical tank with a retention time of 15 minutes and a process flow of 10% of the design water flow for flotation cells and a process flow of 5% of the design meter flow for skim vessels.

$$Q_W = (0.10)\,(5{,}000) + (0.05)\,(5{,}000)$$

$$= 750 \text{ BPD}$$

$$H = \frac{0.7\,(t_r)\,Q_w}{d^2}$$

$$H = \frac{0.8\,(15)\,(750)}{d^2}$$

$$H = \frac{7{,}875}{d^2}$$

Assume various diameters (d) and solve for liquid heights (H). L_{ss} is to equal L_{eff} + 3 ft

Vessel Diameter in.	Effective Length ft	Seam-to-Seam Length ft
30	8.8	11.8
36	6.1	9.1
42	4.5	7.5

A vertical vessel 36 in. × 6 ft would satisfy all design parameters.

CHAPTER

8

Pressure Drop in Piping*

INTRODUCTION

Piping design in production facilities involves the selection of a pipe diameter and a wall thickness that is capable of transporting fluid from one piece of process equipment to another, within the allowable pressure drop and pressure rating restraints imposed by the process. The first step in being able to make these changes is to understand how pressure drops in these lines are calculated. This is discussed in this chapter, while the next chapter discusses the concepts involved in choosing a line size and a pressure rating.

While this chapter emphasizes piping that exists within a facility, the concepts included on pressure drop are equally valid for determining the pressure drop in flowlines, pipelines, gas transmission lines, etc.

This chapter first introduces the basic principles for determining pressure drops in piping and then discusses the flow equations for liquid flow, compressible flow, and two-phase flow. Finally, it shows how to calculate pressure drop in valves and fittings when using the various flow equations. The last part of this chapter includes some example calculations for determining the pressure drop in various types of pipe.

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*Reviewed for the 1998 edition by Eric M. Barron of Paragon Engineering Services, Inc.

BASIC PRINCIPLES

Reynolds Number

The Reynolds number is a dimensionless parameter that relates the ratio of inertial forces to viscous forces. It can be expressed by the following general equation:

$$Re = \frac{\rho D V}{\mu'} \tag{8-1}$$

where Re = Reynolds number
ρ = density, lb/ft^3
D = pipe ID, ft
V = flow velocity, ft/sec
μ' = viscosity, lb/ft-sec

The Reynolds number can be expressed in more convenient terms. For liquids, the equation becomes:

$$Re = 7,738 \frac{(S.G.)\, d\, V}{\mu} \tag{8-2}$$

$$Re = 92.1 \frac{(S.G.)\, Q_l}{d\, \mu} \tag{8-3}$$

where μ = viscosity, cp
d = pipe ID, in.
V = velocity, ft/sec
S.G. = specific gravity of liquid relative to water
Q_l = liquid flow rate, bpd

Derivation of Equations 8-2 and 8-3

μ is in cp, ρ in lb/ft^3, d in inches

$\mu' = \mu/1,488$
$\rho = 62.4$ (S.G.)
$D = d/12$

$$Re = \frac{(62.4)\,(S.G.) d\, V\, (1,488)}{(12)\mu}$$

$$Re = \frac{7,738\, (S.G.) d\, V}{\mu}$$

Q is in ft^3 / sec, A in ft^2

$$V = \frac{Q}{A}$$

$$Q = Q_l \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24hr} \times \frac{hr}{3,600s} = 6.49 \times 10^{-5} Q_l$$

$$A = \frac{\pi D^2}{4} = \frac{\pi d^2}{(4)(144)}$$

$$V = Q_l \frac{(6.49 \times 10^{-5})(4)(144)}{\pi d^2}$$

$$Re = \frac{92.1\,(S.G.)\,Q_l}{d\,\mu}$$

For gases, the equation becomes:

$$Re = 20,100 \frac{Q_g\,S}{d\,\mu} \tag{8-4}$$

where Q_g = gas flow rate, MMscfd
S = specific gravity of gas at standard conditions (air = 1)
d = pipe ID, in.
μ = viscosity, cp

Derivation of Equation 8-4

ρ_g is in lb/ft³, D in ft, μ in cp, μ' in lb/ft-sec

$$Re = \frac{\rho_g\,D\,V_{act}}{\mu'}$$

where V_{act} = actual gas velocity, ft/sec

T is in °R, d in inches, P is psia

$$\rho_g = 0.0764(S) \times \frac{P}{14.7} \times \frac{520}{TZ} = 2.7 \frac{S\,P}{TZ}$$

$D = d/12$

$\mu' = \mu/1,488$

$$V_{act} = \frac{Q_{act}}{A}$$

Q_g is in MMscfd, Q_{act} in ft^3/s

$$Q_{act} = Q_g \times 10^6 \frac{\text{scf}}{\text{MMscf}} \times \frac{\text{day}}{24\text{hr}} \times \frac{\text{hr}}{3{,}600\text{s}} \times \frac{14.7}{P} \times \frac{TZ}{520}$$

$$Q_{act} = 0.327 \frac{TZQ_g}{P}$$

$$A = \frac{\pi d^2}{(4)(144)}$$

$$Re = 20{,}100 \frac{Q_g\, S}{d\, \mu}$$

Flow Regimes

Flow regimes describe the nature of fluid flow. There are two basic flow regimes for flow of a single-phase fluid: laminar flow and turbulent flow. Laminar flow is characterized by little mixing of the flowing fluid and a parabolic velocity profile. Turbulent flow involves complete mixing of the fluid and a more uniform velocity profile. Laminar flow has been shown by experiment to exist at Re < 2,000 and turbulent flow at Re > 4,000. Reynolds numbers between 2,000 and 4,000 are in a transition zone, and thus the flow may be either laminar or turbulent.

Bernoulli's Theorem

It is customary to express the energy contained in a fluid in terms of the potential energy contained in an equivalent height or "head" of a column of the fluid. Using this convention, Bernoulli's theorem breaks down the total energy at a point in terms of

1. The head due to its elevation above an arbitrary datum of zero potential energy.
2. A pressure head due to the potential energy contained in the pressure in the fluid at that point.
3. A velocity head due to the kinetic energy contained within the fluid.

Assuming that no energy is added to the fluid by a pump or compressor, and that the fluid is not performing work as in a steam turbine, the law of conservation of energy requires that the energy at point "2" in the piping system downstream of point "1" must equal the energy at point "1" minus the energy loss to friction and change in elevation. Thus, Bernoulli's theorem may be written:

$$\begin{aligned} &(\text{Elevation Head})_1 && (\text{Elevation Head})_2 \\ &+ (\text{Pressure Head})_1 &= &+ (\text{Pressure Head})_2 \\ &+ (\text{Velocity Head})_1{}^2 && + (\text{Velocity Head})_2{}^2 \\ & && + (\text{Friction Head Loss}) \end{aligned}$$

or

$$Z_1 + \frac{144P_1}{\rho_1} + \frac{V_1^2}{2g} = Z_2 + \frac{144P_2}{\rho_2} + \frac{V_2^2}{2g} + H_L \qquad (8-5)$$

where Z = elevation head, ft
P = pressure, psi
ρ = density, lb/ft^3
V = velocity, ft/sec
g = gravitation constant
H_L = friction head loss, ft

Darcy's Equation

This equation, which is also sometimes called the Weisbach equation or the Darcy-Weisbach equation, states that the friction head loss between two points in a completely filled, circular cross section pipe is proportional to the velocity head and the length of pipe and inversely proportional to the pipe diameter. This can be written:

$$H_L = \frac{fLV^2}{D2g} \qquad (8-6)$$

where L = length of pipe, ft
D = pipe diameter, ft
f = factor of proportionality

Equations 8-5 and 8-6 can be used to calculate the pressure at any point in a piping system if the pressure, flow velocity, pipe diameter, and elevation are known at any other point. Conversely, if the pressure, pipe diameter, and elevations are known at two points, the flow velocity can be calculated.

In most production facility piping systems the head differences due to elevation and velocity changes between two points can be neglected. In this case Equation 8-5 can be reduced to:

$$P_1 - P_2 = \Delta P = \frac{\rho}{144} H_L \tag{8-7}$$

where ΔP = loss in pressure between points 1 and 2, psi

Substituting Equation 8-6 into Equation 8-7, we have:

$$\Delta P = \frac{\rho f L V^2}{144\, D 2 g}$$

Substituting $D = \frac{d}{12}$

where d = pipe diameter, in.

$$\Delta P = \frac{\rho f L\, V^2\, (12)}{(144)\,(2)\,(32.2) d}$$

which reduces to:

$$\Delta P = 0.0013 \frac{f \rho L V^2}{d} \tag{8-8}$$

Moody Friction Factor

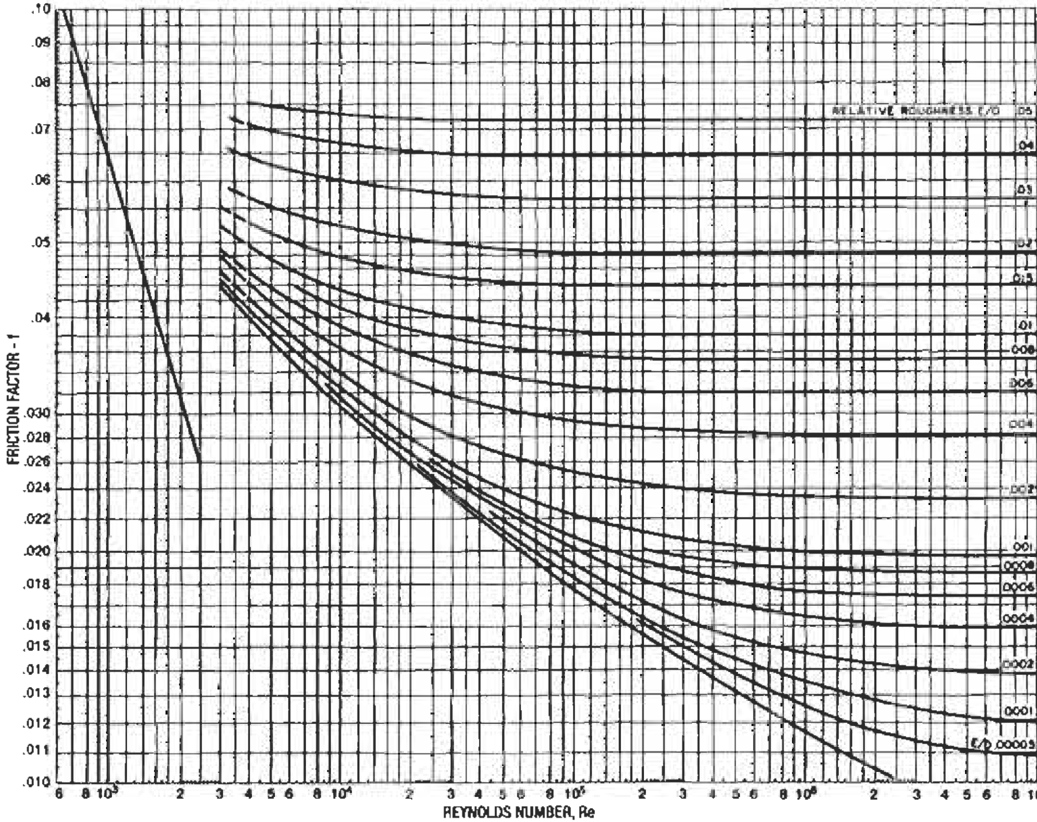
The factor of proportionality in the previous equations is called the Moody friction factor and is determined from the Moody resistance diagram shown in Figure 8-1. The friction factor is sometimes expressed in terms of the Fanning friction factor, which is one-fourth of the Moody friction factor. In some references the Moody friction factor is used, in others, the Fanning friction factor is used. Care must be exercised to avoid inadvertent use of the wrong friction factor.

In general, the friction factor is a function of the Reynolds number, Re, and the relative roughness of the pipe, ε/D. For Laminar flow, f is a function of only the Re:

$$f = \frac{64}{Re} \tag{8-9}$$

For turbulent flow, f is a function of both pipe roughness and the Reynolds number. At high values of Re, f is a function only of ε/D.

Table 8-1 shows the relative roughness for various types of new, clean pipe. These values should be increased by a factor of 2–4 to allow for age and use.



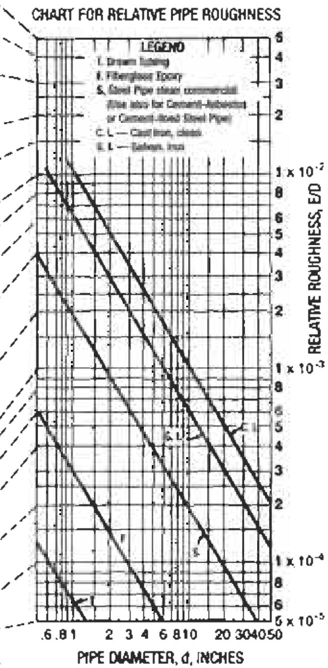


Figure 8-1. Friction factor chart (*courtesy of API RP 14E*).

Table 8-1
Pipe Roughness

Type of Pipe (New, clean condition)	Roughness ε (ft)	Roughness ε (in.)
Unlined Concrete	0.001–0.01	0.012–0.12
Cast Iron—Uncoated	0.00085	0.01
Galvanized Iron	0.0005	0.006
Carbon Steel	0.00015	0.0018
Fiberglass Epoxy	0.000025	0.0003
Drawn Tubing	0.000005	0.00006

Increase by factor of 2–4 to allow for age and use.

FLUID FLOW EQUATIONS

Liquid Flow

$$\Delta P = (11.5 \times 10^{-6}) \frac{fLQ_l^2\,(S.G.)}{d^5} \tag{8-10}$$

where ΔP = pressure drop, psi
f = Moody friction factor, dimensionless
L = length of pipe, ft
Q_l = liquid flow rate, bpd
$S.G.$ = specific gravity of liquid relative to water
d = pipe ID, in.

Derivation of Equation 8-10

ρ is in lb/ft^3, L in ft, V in ft/s, d in inches.

$$\Delta P = 0.0013 \frac{f\rho LV^2}{d}$$

Q in ft^3/s, A in ft^2.

$$V = \frac{Q}{A}$$

Q_l is in bpd.

$$Q = Q_l \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24hr} \times \frac{hr}{3{,}600s} = 6.49 \times 10^{-5} Q_l$$

$$A = \frac{\pi d^2}{(4)(144)}$$

$$\rho = 62.4(S.G.)$$

$$\Delta P = (11.5 \times 10^{-6}) \frac{fLQ_l^2(S.G.)}{d^5}$$

The most common use of Equation 8-10 is to determine a pipe diameter for a given flow rate and allowable pressure drop. However, first, a calculation of Reynolds number (Equation 8-2 or 8-3) to determine the friction factor must be made. Since a Reynolds number depends on the pipe diameter the equation cannot be solved directly. One method to overcome this disadvantage is to assume a typical friction factor of 0.025, solve Equation 8-10 for diameter, compute a Reynolds number, and then compare the assumed friction factor to one read from Figure 8-1. If the two are not sufficiently close, it is possible to iterate the solution until convergence.

Figure 8-2 is a curve that can be used to approximate pressure drop or required pipe diameter. It is based on an assumed friction factor relationship, which can be adjusted to some extent for liquid viscosity.

In an effort to void an iterative calculation, several empirical formulas have been developed. The most common of these is the Hazen-Williams formula, which can be expressed as follows:

$$H_L = 0.00208 \left(\frac{100}{C}\right)^{1.85} \left(\frac{gpm}{d^{4.87}}\right)^{1.85} L \qquad (8-11)$$

$$H_L = 0.015 \frac{Q_l^{1.85} L}{d^{4.87} C^{1.85}} \qquad (8-12)$$

where H_L = head loss due to friction, ft
L = length, ft
C = friction factor constant, dimensionless
= 140 for new steel pipe
= 130 for new cast iron pipe
= 100 for riveted pipe
d = pipe ID, in.
gpm = liquid flow rate, gallons/minute
Q_l = liquid flow rate, bpd

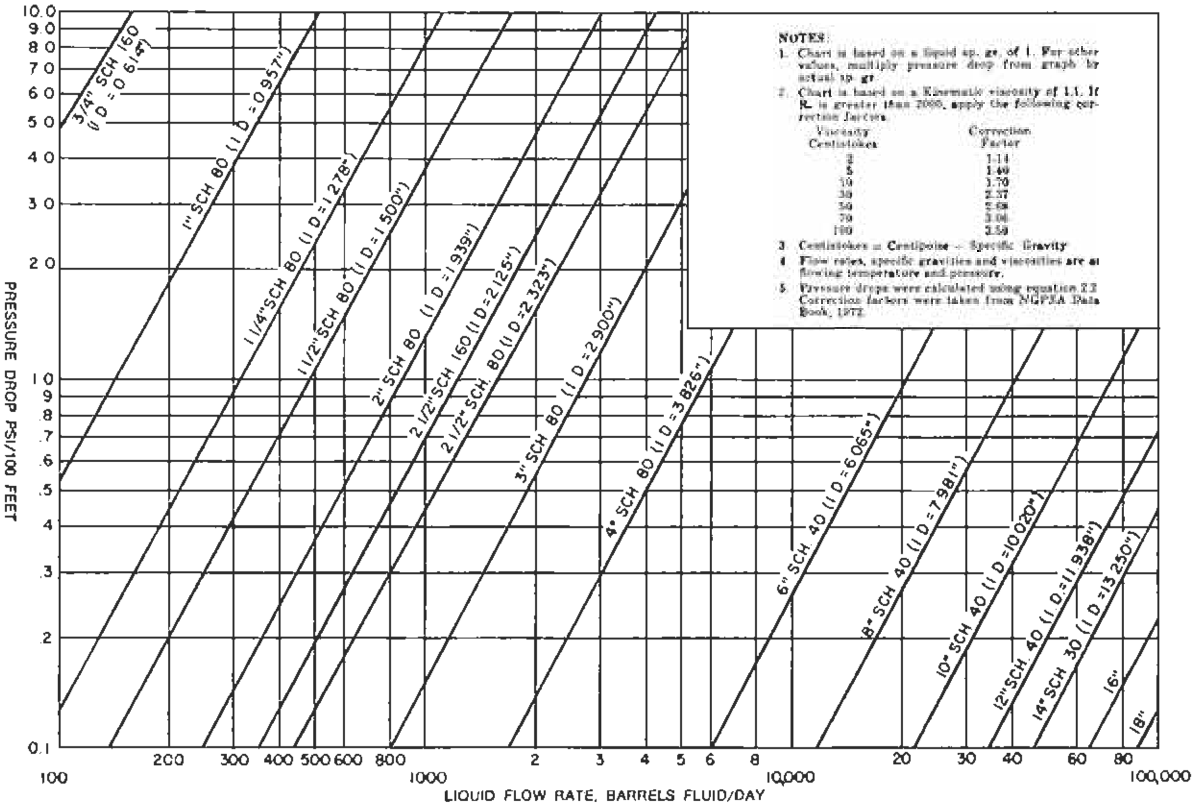


Figure 8-2. Pressure drop in liquid lines (*courtesy of API RP 14E*).

This equation is based on water flowing under turbulent conditions with a viscosity of 1.13 centipoise, which is the case for water at 60°F. Since water viscosity varies appreciably from 32°F to 212°F, the friction factor can decrease or increase as much as 40% between the two temperature extremes.

The Hazen-Williams equation is frequently used for calculating pressure losses and line capacities in water service. The discharge coefficient "C" must be carefully chosen to reflect *both* fluid viscosity and pipe roughness in a used condition. A "C" factor of 90 to 100 in steel pipe is common for most produced liquid problems. Hazen-Williams factor "C" must not be confused with the Moody friction factor "f," as these two factors are not directly related to each other. Typical "C" factors for various types of pipe are shown in Table 8-2.

Table 8-2
Hazen-Williams "C" Factors

Type of Pipe	Values of C RANGE*	AVERAGE**	COMMON***
Welded and Seamless Steel	150–80	140	100
Cement	160–140	150	140
Fibre	—	150	140
Cement-Lined Iron	—	150	140
Bitumastic-Enamel Lined Iron	160–130	148	140
Copper, Brass, Lead, Tin or Glass	150–120	140	130

* *high—best, smooth, well laid; low—poor or corroded*
** *value for good, clean, new pipe*
*** *value used for design purposes.*

Gas Flow

The Darcy equation assumes constant density over the pipe section between the inlet and outlet points. While this assumption is valid for liquids, it is incorrect for pipelines flowing gases, where density is a strong function of pressure and temperature. As the gas flows through the pipe it expands due to the drop in pressure and thus tends to decrease in density. At the same time, if heat is not added to the system, the gas will cool, causing the gas to tend to increase in density. In control valves, where the change in pressure is near instantaneous, and thus no heat is added to the system, the expansion can be considered adiabatic. In pipe flow, however, the pressure drop is gradual and there is sufficient pipe surface area between the gas and the surrounding medium to add heat to the gas and

thus keep the gas at constant temperature. In such a case the gas can be considered to undergo an isothermal expansion.

On occasion, where the gas temperature is significantly different from ambient, the assumption of isothermal (constant temperature) flow is not valid. In these instances greater accuracy can be obtained by breaking the line up into short segments that correspond to only small temperature changes.

The general isothermal equation for the expansion of gas can be given by:

$$w^2 = \left[\frac{144 g A^2}{\bar{V}_1 \left(\frac{fL}{D} + 2 \log_e \frac{P_1}{P_2} \right)} \right] \left[\frac{(P_1)^2 - (P_2)^2}{P_1} \right] \qquad (8-13)$$

where w = rate of flow, lb/sec
g = acceleration of gravity, ft/sec^2
A = cross-sectional area of pipe, ft^2
$\bar{V}_1$ = specific volume of gas at upstream conditions, ft^3/lb
f = friction factor
L = length, ft
D = diameter of pipe, ft
P_1 = upstream pressure, psia
P_2 = downstream pressure, psia

This equation assumes that

1. No work is performed between points 1 and 2, i.e., there are no compressors or expanders and no elevation changes.
2. The gas is flowing under steady state conditions, i.e., no acceleration changes.
3. The Moody friction factor, f, is constant as a function of length. There is some change due to a change in Reynolds number, but this is quite small.

For practical pipeline purposes,

$2 \log_e \frac{P_1}{P_2} << \frac{fL}{D}$ and can be ignored.

With this assumption and substituting in Equation 8-13 for practical oil field units:

$$P_1^2 - P_2^2 = 25.1 \frac{S Q_g^2 Z T_1 f L}{d^5} \qquad (8-14)$$

where P_1 = upstream pressure, psia
P_2 = downstream pressure, psia
S = gas specific gravity at standard conditions
Q_g = gas flow rate, MMscfd
Z = compressibility factor for gas
T_1 = flowing temperature, °R
f = Moody friction factor, dimensionless
d = pipe ID, in.

Derivation of Equation 8-14

$$w^2 = \left[\frac{144 g\, A^2}{\overline{V}_1\left(\frac{fL}{D} + 2\log_e \frac{P_1}{P_2}\right)}\right]\left[\frac{(P_1)^2 - (P_2)^2}{P_1}\right]$$

Q_g is in MMscfd, S is at STP, w in lb/sec.

$$w = Q_g \times \frac{1,000,000}{(24)(3,600)} \times 0.0764\,(S) = 0.884\,(S)\,Q_g$$

d is in inches, A in ft^2.

$D = d/12$

$$A = \frac{\pi d^2}{(4)(144)} = 0.00545\, d^2$$

T_1 is in °R, P_1 in psia, ρ in lb / ft^3

$$\frac{P_1 \overline{V}_1}{Z_1 T_1} = \frac{P_s V_s}{Z_s T_s}$$

At standard conditions $P_s = 14.7$, $\overline{V}_s = \frac{1}{\rho} = \frac{1}{0.0764(S)}$, $T_s = 520$

$Z_s \cong 1.0$

$$\overline{V}_1 = \frac{1}{0.0764(S)} \times \frac{14.7}{P_1} \times \frac{ZT_1}{520}$$

$$\overline{V}_1 = 0.370\,\frac{ZT_1}{SP_1}$$

$$2 \log_e \frac{P_1}{P_2} \cong 0$$

$$[0.883\, S\, Q_g]^2 = \frac{(144)(32.2)(0.00545)^2 d^4 S P_1 d}{(0.370)\, T_1 Z f L\, (12)} \left[\frac{P_1^2 - P_2^2}{P_1} \right]$$

$$P_1^2 - P_2^2 = 25.1 \frac{S Q_g^2\, Z\, T_1 f\, L}{d^5}$$

The "Z" factor will change slightly from point 1 to point 2. It is usually assumed to be constant and is chosen for an "average" pressure of:

$$2/3 \left[P_1 + P_2 - \frac{P_1 P_2}{P_1 + P_2} \right] \qquad (8-15)$$

Rearranging Equation 8-14 and solving for Q_g we have:

$$Q_g = 0.199 \left[\frac{d^5 (P_1{}^2 - P_2{}^2)}{Z\, T_1\, f\, L\, S} \right]^{1/2} \qquad (8-16)$$

An approximation of Equation 8-14 can be made when the change in pressure is less than 10% of the inlet pressure. If this is true we can make the assumption:

$$P_1{}^2 - P_2{}^2 \cong 2P_1\,(P_1 - P_2)$$

Substituting into Equation 8-14 we have:

$$\Delta P = 12.6 \left[\frac{S\, Q_g^2 Z T_1 f L}{P_1 d^5} \right] \qquad (8-17)$$

As was the case for liquid flow, in order to solve any of these equations for a pipe diameter to handle a given flow and pressure drop, it is necessary to first guess the diameter and then compute a Reynolds number to determine the friction factor. Once the friction factor is known, a pipe diameter can be calculated and compared against the assumed number. The process can be iterated until convergence.

Several empirical gas flow equations have been developed. These equations are patterned after the general flow equation (Equation 8-16), but make certain assumptions so as to avoid solving for the Moody friction factor. The three most common gas flow equations are described in the following sections.

Weymouth Equation

This equation is based on measurements of compressed air flowing in pipes ranging from 0.8 in. to 11.8 in. in the range of the Moody diagram where the ε/d curves are horizontal (i.e., high Reynolds number). In this range the Moody friction factor is independent of the Reynolds number and dependent upon the relative roughness. For a given absolute roughness, ε, the friction factor is merely a function of diameter. For steel pipe the Weymouth data indicate:

$$f = \frac{0.032}{d^{1/3}} \tag{8-18}$$

Substituting this into Equation 8-16, the Weymouth equation expressed in practical oil field units is:

$$Q_g = 1.11\, d^{2.67} \left[\frac{P_1^2 - P_2^2}{LSZT_1} \right]^{1/2} \tag{8-19}$$

where Q_g = flow rate, MMscfd
d = pipe ID, in.
P_1 and P_2 = pressure at points 1 and 2 respectively, psia
L = length of pipe, ft
S = specific gravity of gas at standard conditions
T_1 = temperature of gas at inlet, °R
Z = compressibility factor of gas

Assuming a temperature of 520°R, a compressibility of 1.0 and a specific gravity of 0.6 the Weymouth equation can also be written:

$$Q'_g = 865\, d^{2.67} \left[\frac{P_1^2 - P_2^2}{L_m} \right]^{1/2} \tag{8-20}$$

where Q'_g = flow rate, scfd
L_m = pipe length, miles

This is the form of the equation given in the Gas Processors Suppliers Association (GPSA) Engineering Data Book. The correction factors for gravity and temperature are merely ratios of square roots of the assumed values divided by the actual values.

It is important to remember the assumptions used in deriving this equation and when they are appropriate. Short lengths of pipe with high pressure drops are likely to be in turbulent flow and thus the assump-

tions made by Weymouth are appropriate. Industry experience indicates that Weymouth's equation is suitable for most piping within the production facility. However, the friction factor used by Weymouth is generally too low for large diameter or low velocity lines where the flow regime is more properly characterized by the sloped portion of the Moody diagram.

Panhandle Equation

This equation was intended to reflect the flow of gas through smooth pipes and is a reasonable approximation of partially turbulent flow behavior. The friction factor can be represented by a straight line of constant negative slope in the moderate Reynolds number region of the Moody diagram.

A straight line on the Moody diagram would be expressed:

$$\log f = n \log Re + \log C \tag{8-21}$$

or

$$f = \frac{C}{R_e^n} \tag{8-22}$$

The Panhandle A equation applies to Reynolds numbers in the 5×10^6 to 11×10^6 range and assumes n = 0.146. The Panhandle B equation assumes more fully developed turbulent flow (greater Reynolds number) and assumes a lower slope of n = 0.039.

Using this assumption and assuming a constant viscosity for the gas, the Panhandle A equation can be written:

$$Q_g = 0.020\, E \left[\frac{P_1^{\,2} - P_2^{\,2}}{S^{0.853}\, ZT_1L_m} \right]^{0.059} d^{2.62} \tag{8-23}$$

Panhandle B can be written:

$$Q_g = 0.028\, E \left[\frac{P_1^{\,2} - P_2^{\,2}}{S^{0.961}\, ZT_1L_m} \right]^{0.51} d^{2.53} \tag{8-24}$$

where E = efficiency factor
= 1.0 for brand new pipe
= 0.95 for good operating conditions
= 0.92 for average operating conditions
= 0.85 for unfavorable operating conditions

Derivation of Equations 8-23 and 8-24

$$Q_g = 0.199\left[\frac{d^5(P_1^2 - P_2^2)}{Z\,T_1\,f\,L\,S}\right]^{1/2}$$

$$Re = 20{,}100\,\frac{Q_g S}{d\mu}$$

$$f = \frac{C}{Re^n}$$

$$C' = C\left(\frac{\mu}{20{,}100}\right)^n$$

$$f = C'\left(\frac{d}{Q_g\,(S)}\right)^n$$

$$L = 5{,}280\,L_m$$

For Panhandle A assume:

$n = 0.146$

$C' = 0.010$

For Panhandle B assume:

$n = 0.039$

$C' = 0.008$

In practice, the Panhandle equations are commonly used for large diameter, long pipelines where the Reynolds number is on the straight line portion of the Moody diagram. It can be seen that neither the Weymouth nor the Panhandle equations represent a "conservative" assumption. If the Weymouth formula is assumed, and the flow is a moderate Reynolds number, the friction factor will in reality be higher than assumed (the sloped line portion is higher than the horizontal portion of the Moody curve), and the actual pressure drop will be higher than calculated. If the Panhandle B formula is used and the flow is actually in a high Reynolds number, the friction factor will in reality be higher than assumed (the equation assumes the friction factor con-

tinues to decline with increased Reynolds number beyond the horizontal portion of the curve), and the actual pressure drop will be higher than calculated.

Spitzglass Equation

This equation is used for near-atmospheric pressure lines. It is derived directly from Equation 8-16 by making the following assumptions:

1. $f = \left(1 + \frac{3.6}{d} + 0.03d\right)\left(\frac{1}{100}\right)$
2. $T = 520°R$
3. $P_1 = 15$ psi
4. $Z = 1.0$
5. $\Delta P < 10\%$ of P_1

With these assumptions, and expressing pressure drop in terms of inches of water, the Spitzglass equation can be written:

$$Q_g = 0.09\left[\frac{\Delta h_w\, d^5}{S\,L\left(1 + \frac{3.6}{d} + 0.03d\right)}\right]^{1/2} \qquad (8-25)$$

where Δh_w = pressure loss, inches of water

Derivation of Equation 8-25

$$Q_g = 0.199\left[\frac{d^5(P_1^2 - P_2^2)}{Z\,T_1\,f\,L\,S}\right]^{1/2}$$

$$\Delta P < 10\%\ P_1$$

$$\Delta P = 12.6\,\frac{S\,Q_g^2 Z T_1 f L}{P_1 d^5}$$

$$Q_g = 0.282\left[\frac{(\Delta P)\,P_1\,d^5}{S\,Z\,T_1\,f\,L}\right]^{1/2}$$

$P_1 = 15,\ Z = 1.0,\ T_1 = 520$

$$\Delta h_w = \frac{(\Delta P)(144)(12)}{62.4}$$

$$\Delta P = 0.036\,\Delta h_w$$

$$Q_g = 0.282\left[\frac{0.036\,\Delta h_w\,(15)\,d^5}{S(1.0)(520)\,f\,L}\right]^{1/2}$$

$$Q_g = 0.009\left[\frac{\Delta h_w\,d^5}{S\,f\,L}\right]^{1/2}$$

$$f = \left(1 + \frac{3.6}{d} + 0.03d\right)\left(\frac{1}{100}\right)$$

$$Q_g = 0.09\left[\frac{\Delta h_w\,d^5}{S\,L\left(1 + \frac{3.6}{d} + 0.03d\right)}\right]^{1/2}$$

Application of Gas Flow Equations

The Weymouth and Spitzglass equations both assume that the friction factor is merely a function of pipe diameter. Figure 8-3 compares the friction factors calculated from these equations with the factor indicated by the horizontal line of the Moody diagram for two different absolute roughnesses.

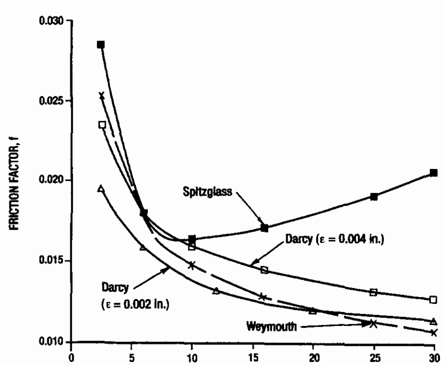


Figure 8-3. Friction factor vs. pipe diameter for three correlations.

In the small pipe diameter range (3–6 in.) all curves tend to yield identical results. For large diameter pipe (10 in. and larger) the Spitzglass equation becomes overly conservative. The curve is going in the wrong direction, thus the form of the equation must be wrong. The Weymouth equation tends to become under-conservative with pipe greater than 20 inches. Its slope is greater than the general flow equation with $\varepsilon = 0.002$ inch. This is merely a result of the way in which the Spitzglass and Weymouth equations represent the Moody diagram.

The empirical gas flow equations use various coefficients and exponents to account for efficiency and friction factor. These equations represent the flow condition upon which they were derived, but may not be accurate under different conditions. Unfortunately, these equations are often used as if they were universally applicable. The following guidelines are recommended in the use of gas flow equations:

1. The general gas flow equation is recommended for most general usage. If it is inconvenient to use the iterative procedure of the general equation and it is not known whether the Weymouth or the Panhandle equations are applicable, compute the results using *both* Weymouth and Panhandle equations and use the *higher* calculated pressure drop.
2. Use the Weymouth equation only for small-diameter, short-run pipe within the production facility where the Reynolds number is expected to be high.
3. Use the Panhandle equations only for large-diameter, long-run pipelines where the Reynolds number is expected to be moderate.
4. Use the Spitzglass equation for low pressure vent lines less than 12-inches in diameter.
5. When using gas flow equations for old pipe, attempt to derive the proper efficiency factor through field tests. Buildup of scale, corrosion, liquids, paraffin, etc. can have a large effect on gas flow efficiency.

Two-Phase Flow

Two-phase flow of liquid and gas is a very complex physical process. Even when the best existing correlations for pressure drop and liquid holdup are used, predictions may be in error as much as ±20%. Nevertheless, as gas exploration and production have moved into remote offshore, arctic, and desert areas, the number of two-phase pipelines has increased.

To determine whether two-phase flow will exist in a pipeline, the expected flowing pressure and temperature ranges in the line must be plotted on a phase diagram for the fluid. Figure 8-4 shows that composition B will flow as a single-phase fluid as it enters the pipeline. However, as the pressure drops it becomes a two-phase mixture through part of the pipeline. On the other hand, composition A will flow as a single-phase (dense fluid or gas) through the entire length of the line. Composition C will flow as a liquid throughout the entire length of the line.

In most production situations the fluid coming out of the well bore will be in two-phase flow. Once an initial separation is made, the gas coming off the separator can be considered to be single-phase gas flow even though it will have some entrained liquids. The liquid coming off the separator is assumed to be in single-phase liquid flow even though it will contain some gas after it has taken a pressure drop through a liquid control valve.

Other than well flowlines, the most common two-phase pipelines exist in remote locations, especially offshore, where gas and oil that have been separated and metered are combined for flow in a common line to a central separation facility.

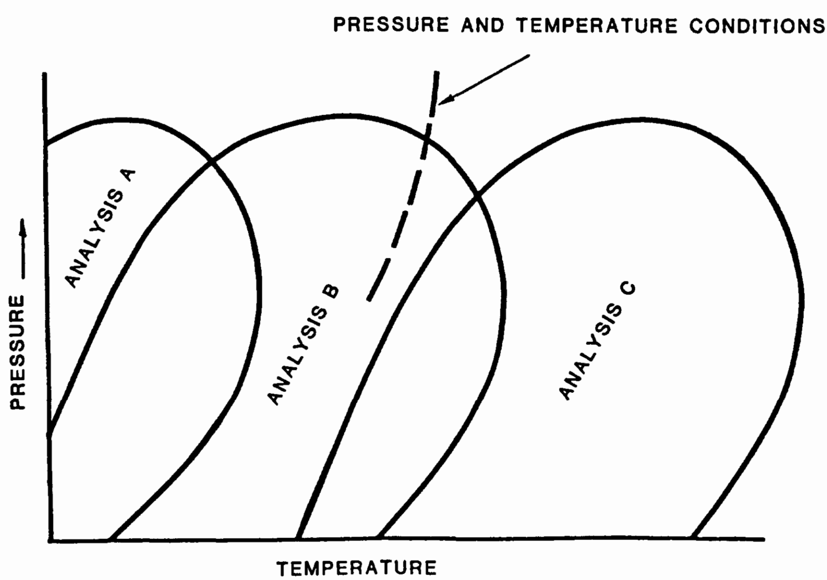


Figure 8-4. Typical hydrocarbon phase behavior.

Horizontal Flow

When a gas-liquid mixture enters a pipeline, the two phases tend to separate with the heavier liquid gravitating to the bottom. Figure 8-5 shows typical flow patterns in horizontal two-phase pipe flow. The type of flow pattern depends primarily on the superficial velocities as well as the system geometry and physical properties of the mixture. At very low gas-liquid ratios, the gas tends to from small *bubbles* that rise to the top of the pipe. As the gas-liquid ratio increases, the bubbles become larger and eventually combine to form *plugs.* Further increases in the gas-liquid ratio cause the plugs to become longer, until finally the gas and liquid phases flow in separate layers; this is *stratified* flow. As the gas flow rate is increased, the gas-liquid interface in stratified flow becomes *wavy.* These waves become higher with increasing gas-liquid ratios, until the crest of the waves touches the top of the pipe to form *slugs* of liquid which are pushed along by the gas behind them. These slugs can be several hundred feet long in some cases. Further increases in the gas-liquid ratio may impart a centrifugal motion to the liquid an result in *annular* flow. At extremely high gas-liquid ratios, the liquid is *dispersed* into the flowing gas stream.

Figure 8-6 shows how the flow regime for horizontal pipes depends primarily on the superficial gas and liquid flow rates. Experience has shown that generally flow regime maps such s Figure 8-6 are not very accurate, but they can be used as qualitative guides.

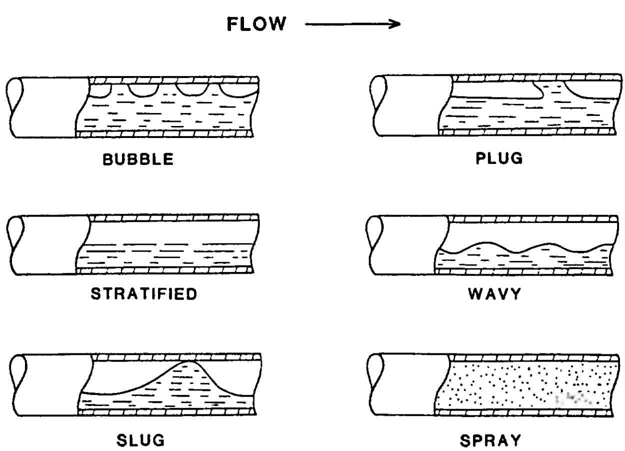


Figure 8-5. Two-phase flow patterns in horizontal flow.

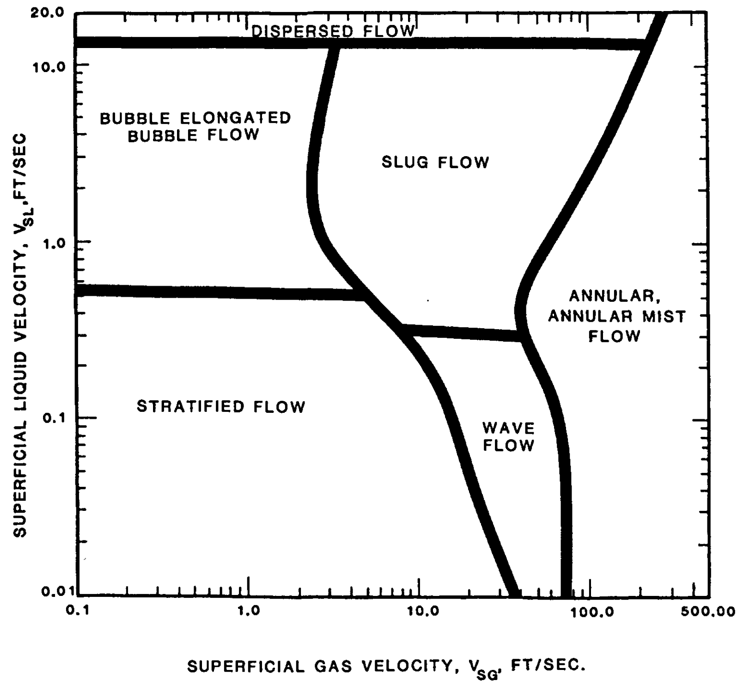


Figure 8-6. Flow regime for horizontal pipes. (Reprinted with permission from *International Journal of Multiphase Flow,* Vol. 1, J. M. Mandhane, G. A. Gregory, and K. Aziz, "A Flow Pattern Map for Gas-Liquid Flow in Horizontal Pipes," 1974, Pergamon Press, Ltd.)

In most two-phase flow lines in the field, slug flow is predominant in level and uphill lines. In downhill lines, stratified flow is predominant. However, if the slope of the downhill line is not very steep and the gas velocity is high, slug flow may be observed. The criterion for transition from stratified to slug flow in downhill lines is not well defined.

Vertical Flow

The two-phase flow patterns in vertical flow are somewhat different from those occurring in horizontal or slightly inclined flow. Vertical two-phase flow geometrics can be classified as bubble, slug-annular, transition, and annular-mist, depending on the gas-liquid ratio. All four flow regimes could conceivably exist in the same pipe. One example is a deep well producing light oil from a reservoir that is near its bubble point. At the bottom of the hole, with little free gas present, flow would be in the

bubble regime. As the fluid moves up the well, the other regimes would be encountered because gas continually comes out of solution as the pressure continually decreases. Normally flow is in the slug regime and rarely in mist, except for condensate reservoirs or steam-stimulated wells. The different flow regimes are shown in Figure 8-7. Figure 8-8 gives approximate flow regimes as a function of superficial gas and liquid flow rates. These flow regimes are described below:

1. *Bubble Flow:* The gas-liquid ratio is small. The gas is present as small bubbles, randomly distributed, whose diameters also vary randomly. The bubbles move at different velocities depending upon their respective diameters. The liquid moves up the pipe at a fairly uniform velocity, and except for its density, the gas phase has little effect on the pressure gradient.
2. *Slug Flow:* In this regime the gas phase is more pronounced. Although the liquid phase is still continuous, the gas bubbles coalesce and form stable bubbles of approximately the same size and shape, which are nearly the diameter of the pipe. They are separated by slugs of liquid. The bubble velocity is greater than that of the liquid and can be predicted in relation to the velocity of the liquid slug. There is a film of liquid around the gas bubble. The liquid velocity is not constant; whereas the liquid slug always moves upward (in the direction of bulk flow), the liquid in the film may move upward, but possibly at a lower velocity, or it may even move downward. These varying liquid veloci-

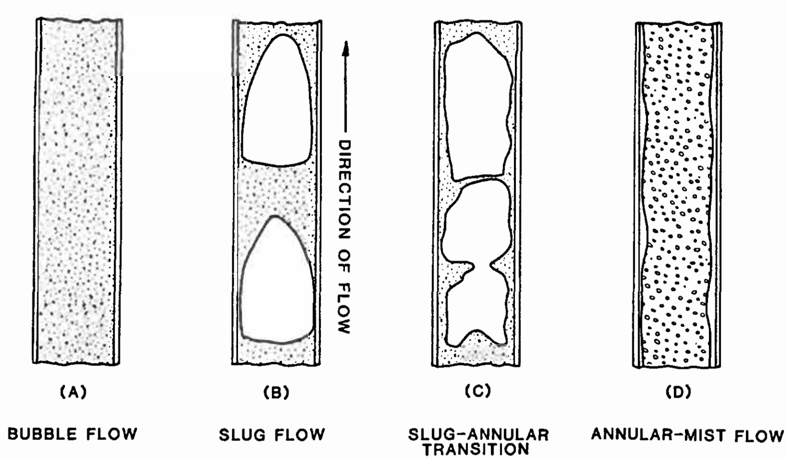


Figure 8-7. Two-phase flow patterns in vertical flow.

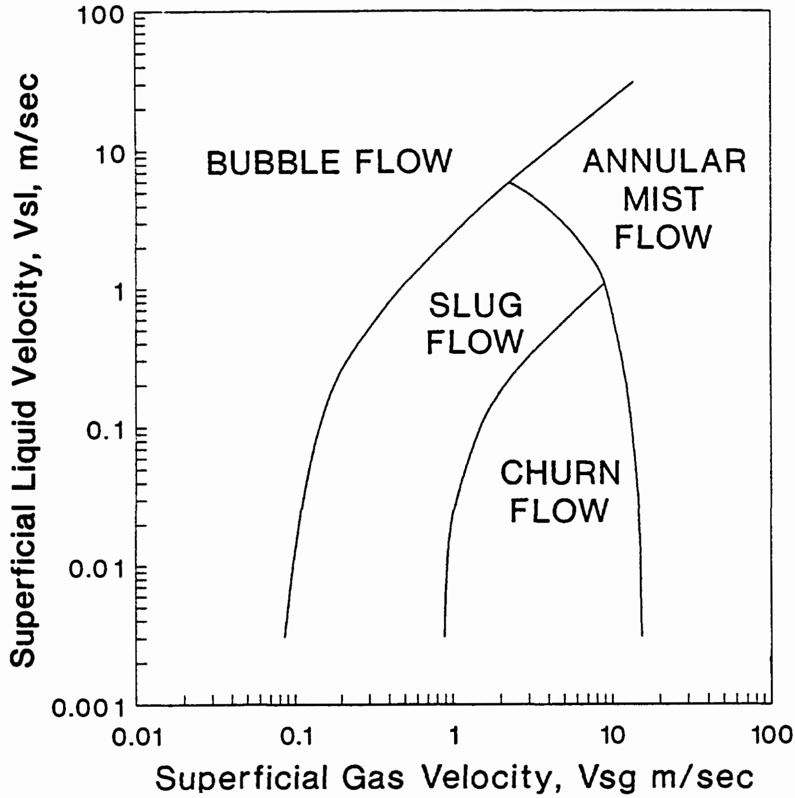


Figure 8-8. Vertical multiphase flow map (from *AIChE J.*, Y. Yaitel, D. Barnea, and A. E. Duckler, "Modeling Flow Pattern Transitions for Steady Upward Gas-Liquid Flow in Vertical Tubes," May 1980).

ties not only result in varying wall friction losses, but also result in liquid holdup, which influences flowing density. At higher flow velocities, liquid can even be entrained in the gas bubbles. Both the gas and liquid phases have significant effects on pressure gradient.

3. *Transition Flow:* The change from a continuous liquid phase to a continuous gas phase occurs in this region. The liquid slug between the bubbles virtually disappears, and a significant amount of liquid becomes entrained in the gas phase. In this case, although the effects of the liquid are significant, the gas phase is predominant. Transition flow is also known as "churn flow."

4. *Annular-Mist Flow:* The gas phase is continuous. The bulk of the liquid is entrained and carried in the gas phase. A film of liquid wets

the pipe wall, but its effects are secondary. The gas phase is the controlling factor.

Pressure Drop

Pressure drop in two-phase flow is the sum of the pressure drop due to acceleration, friction losses, and elevation changes.

In most pipelines, the pressure loss due to acceleration is small. Pressure drop due to friction is typically several times larger in two-phase flow than the sum of the pressure drops of the equivalent two single phases. The additional frictional pressure drop in two-phase flow is attributed to irreversible energy transfer between phases at the interface and to the reduced cross-sectional area available for the flow to each phase.

Pressure drop due to elevation changes is also significant in two-phase flow. In an uphill line, the pressure drop due to elevation change is merely the average density of the two-phase mixture in the uphill line multiplied by the change in elevation. Since the average density depends on the liquid holdup, the static head disadvantage in an uphill line also depends on the average liquid holdup for the segment.

In cross-country lines that consist of a number of uphill and downhill segments, the worst case for pressure drop occurs at low gas flow rates, where each uphill segment fills with liquid. That is, in these segments the liquid holdup approaches the volume of the pipe and the gas is in bubble flow. When this happens, the pressure drop in each uphill segment is given by:

$$\Delta P_z \cong 0.43\,(\text{S.G.})\,\Delta Z_n \qquad (8\text{–}26)$$

where ΔP_z = pressure drop due to elevation increase in the segment, psi
ΔZ_n = increase in elevation for segment n, ft
S.G. = specific gravity of the fluid in the segment relative to water

Derivation of Equation 8-26

ΔP_z in psi, ΔZ_n in ft, ρ in lb/ft^3,

$$\Delta P_z = \frac{\rho\,\Delta Z_n}{144}$$

$$\rho = 62.4\,(\text{S.G.})$$

$$\Delta P_z = \frac{62.4}{144}\,(S.G.)\Delta Z_n$$

$$\Delta P_z = 0.43\,(S.G.)\,\Delta Z_n$$

In downhill lines, flow is normally stratified, and the liquid flows as fast or faster than the gas. The depth of the liquid layer adjusts to the critical depth at which the static head advantage due to the downhill run balances the frictional losses and the pressure loss can be considered to be zero. If slug flow exists in the downhill segment due to a slight slope and/or high gas velocity, the static head advantage will generally be less than the frictional pressure drop, resulting in a net pressure loss. In cross-country lines the static head advantage is normally neglected, and in hand calculations no pressure recovery due to the decrease in elevation is considered.

The net effect of a pressure loss due to an increase in elevation on each uphill segment and no pressure recovery on each downhill segment can be quite severe. In a single-phase line only the net change in elevation from the beginning of the line to the end of the line need be considered. In a two-phase line Equation 8-26 states the pressure lost in each uphill segment, but this is not balanced by a pressure gain in each downhill segment. Thus, the pressure lost due to elevation changes is the sum of the pressure lost in *each* uphill segment. It is possible in hilly terrain to have a line that flows from a high point to a lower elevation, but which, because it crosses a valley, will still have a *loss* in pressure due to elevation changes.

There are numerous computer programs available that calculate pressure drops in two-phase flow. If these programs are available, by all means, use them. However, in the absence of these programs, hand calculation methods have been developed to calculate pressure drops in two-phase flow. One equation presented in the American Petroleum Institute's Recommended Practice API RP 14E entitled "Design and Installation of Offshore Production Piping Systems," is:

$$\Delta P = \frac{3.4 \times 10^{-6} fLW^2}{\rho_m\, d^5} \qquad (8-27)$$

where L = length, ft
W = rate of flow of liquid and vapor, lb/hr
ρ_m = density of the mixture, lb/ft^3
d = pipe ID, in.

For $f = 0.015$

$$\Delta P = \frac{5 \times 10^{-8} L\, W^2}{\rho_m\, d^5} \tag{8-28}$$

This equation is derived from the general equation for isothermal flow by making the following assumptions:

1. ΔP is less than 10% of inlet pressure
2. Bubble or mist flow exists
3. No elevation changes

Derivation of Equations 8-27 and 8-28

$$w^2 = \left[\frac{(144) g\, A^2 D}{\overline{V}_1 f L}\right]\left[\frac{(P_1)^2 - (P_2)^2}{P_1}\right]$$

D is in ft, $\overline{V}$ in ft^3/lb, L in ft, A in ft^2, w in lb/sec.

$$\text{For } \frac{\Delta P}{P_1} < 10\%,\ P_1^2 - P_2^2 \approx 2 P_1 \Delta P$$

$$\Delta P = \left[\frac{\overline{V}\, f\, L}{(144) g\, A^2 D}\right]\frac{w^2}{2}$$

$g = 32.2$

ρ_m is in lb/ft^3, W in lb/hr

$$\overline{V} = \frac{1}{\rho_m}$$

$A = 0.00545 d^2$

$D = d/12$

$$w = \frac{\dot{W}}{3{,}600}$$

$$\Delta P = \frac{3.4 \times 10^{-6}\, f\, L\, W^2}{\rho_m\, d^5}$$

$f = 0.015$

$$\Delta P = \frac{5 \times 10^{-8}\, L\, W^2}{\rho_m\, d^5}$$

The rate of flow of the mixture to use in this equation can be calculated as follows:

$$W = 3{,}180\ Q_g\ S + 14.6\ Q_l\ (S.G.) \qquad (8\text{–}29)$$

where Q_g = gas flow rate, MMscfd
Q_l = liquid flow rate, bpd
S = specific gravity of gas at standard conditions (air = 1)
$(S.G.)$ = specific gravity of liquid relative to water

Derivation of Equation 8-29

L is liquid flow rate in lb/hr, Q_l in bpd

$$L = Q_l\ \frac{\text{barrel}}{\text{day}} \times 5.61\ \frac{\text{ft}^3}{\text{barrel}} \times \frac{\text{day}}{24\text{hr}} \times 62.4\ \text{S.G.}\ \frac{\text{lb}}{\text{ft}^3}$$

$$L = 14.6\ Q_l\ (S.G.)$$

G is gas flow rate in lb/hr, Q_g in MMscfd,

$$G = Q_g\ \frac{\text{MMscf}}{\text{day}} \times 1{,}000{,}000\ \frac{\text{scf}}{\text{MM}} \times 0.0764\ (S)\ \frac{\text{lb}}{\text{scf}} \times \frac{1}{24}\ \frac{\text{day}}{\text{hr}}$$

$$G = 3{,}180\ Q_g\ S$$

$$W = G + L$$

$$W = 3{,}180\ Q_g\ S + 14.6\ Q_l\ (S.G.)$$

The density of the mixture to use in Equations 8-27 and 8-28 is given by:

$$\rho_m = \frac{12{,}409\ (S.G.)\ P + 2.7\ R\ S\ P}{198.7\ P + R\ T\ Z} \qquad (8-30)$$

where P = operating pressure, psia
R = gas/liquid ratio, ft³/bbl
T = operating temperature, °R
Z = gas compressibility factor

Derivation of Equation 8-30

g is gas flow rate in ft³/sec, Q_g in MMscfd,

$$g = Q_g\ \frac{1{,}000{,}000\ \text{scf}}{\text{MM}} \times \frac{\text{day}}{24\ \text{hr}} \times \frac{\text{hr}}{3{,}600\text{s}} \times \frac{14.7}{P} \times \frac{TZ}{520}$$

$$g = 0.327\ \frac{Q_g\ T\ Z}{P}$$

l is in ft³/sec, Q_l in bpd,

$$l = Q_l \times 5.61 \frac{ft^3}{barrel} \times \frac{day}{24hr} \times \frac{hr}{3{,}600s}$$

$$l = 6.49 \times 10^{-5}\ Q_l$$

ρ_m is in lb/ft³, W is in lb/hr,

$$W = 3{,}180\ Q_g\ S + 14.6\ Q_l\ (S.G.)$$

$$\rho_m = \frac{W}{3{,}600\ (l + g)}$$

$$\rho_m = \frac{3{,}180\ Q_g\ S + 14.6\ Q_l\ (S.G.)}{3{,}600\left(6.49 \times 10^{-5}\ Q_l + 0.327\ \frac{Q_g\ T\ Z}{P}\right)}$$

$$R = \frac{1{,}000{,}000\ Q_g}{Q_l}$$

$$\rho_m = \frac{3{,}180\ Q_g\ S + \frac{(14.6)\ (1{,}000{,}000)\ Q_g\ (S.G.)}{R}}{3{,}600\left(\frac{6.49 \times 10^{-5} \times 10^6 \times Q_g}{R} + 0.327\ \frac{Q_g\ T\ Z}{P}\right)}$$

Factor out Q_g, multiply top and bottom by $R \times P$ and rearrange

$$\rho_m = \frac{14.6 \times 10^6\ (S.G.)\ P + 3{,}180\ R\ S\ P}{3{,}600\ (64.9\ P + 0.327\ R\ T\ Z)}$$

Divide top and bottom by 1,177

$$\rho_m = \frac{12{,}409\ (S.G.)\ P + 2.7\ R\ S\ P}{198.7\ P + R\ T\ Z}$$

HEAD LOSS IN VALVES AND PIPE FITTINGS

In many piping problems, especially those associated with offshore production facilities where space limitation are important, the pressure drop through valves, bends, tees, enlargements, contractions, etc. becomes very important. The three most common ways of handling these

additional pressure drops are by use of resistance coefficients, flow coefficients, and equivalent lengths.

Resistance Coefficients

Darcy's equation can be rewritten as:

$$H_L = K_r \frac{V^2}{2g} \tag{8-31}$$

where K_r = resistance coefficient
= fL/D

Although K_r depends on the Reynolds number and surface roughness, as well as on the geometry of the elbow or couplings, this dependence is usually neglected. Approximate values of K_r are given in Table 8-3 for various pipe fittings. Figures 8-9 and 8-10 show resistance coefficients for sudden contractions and expansions, and for entrances and exits. Note that the friction factor (f_t) in Figure 8-9 assumes fully turbulent flow through standard tees and elbows and is the same factor as stated previously in the text.

Table 8-3
Resistance Coefficients for Pipe Fittings

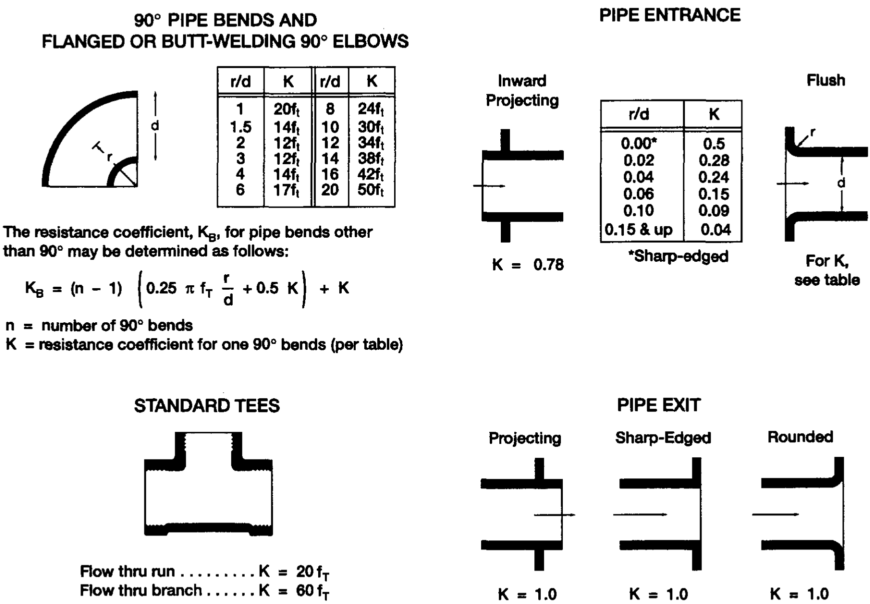
Global Valve, wide open	10.0
Angle Valve, wide open	5.0
Gate Valve, wide open	0.2
Gate Valve, half open	5.6
Return Blend	2.2
Tee	1.8
90° Elbow	0.9
45° Elbow	0.4

The total head loss for the entire piping system can be determined from the following equation:

$$H_L = \sum K_r \frac{V^2}{2g} \tag{8-32}$$

Flow Coefficient

The pressure drop characteristics of control valves are often expressed in terms of C_v, the flow coefficient. The flow coefficient is measured experimentally for each valve or fitting and is equal to the flow of water,



r/d	K	r/d	K
1	$20f_t$	8	$24f_t$
1.5	$14f_t$	10	$30f_t$
2	$12f_t$	12	$34f_t$
3	$12f_t$	14	$38f_t$
4	$14f_t$	16	$42f_t$
6	$17f_t$	20	$50f_t$

The resistance coefficient, K_B, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\ \pi\ f_T\ \frac{r}{d} + 0.5\ K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bends (per table)

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

*Sharp-edged

Flow thru run . . . . $K = 20 f_T$
Flow thru branch . . . . . . $K = 60 f_T$

Figure 8-9. Representative resistance coefficients (K) for fittings (*courtesy of Crane Technical Paper 410*).

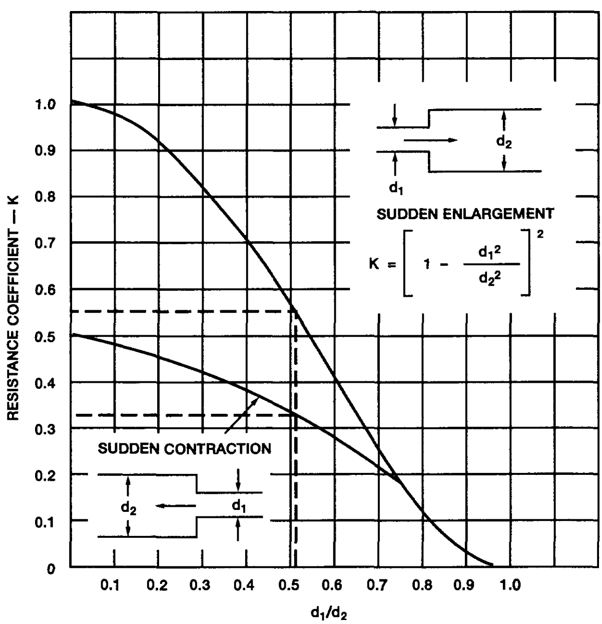


Figure 8-10. Resistance in pipe due to sudden enlargements and contractions.

in gpm, at 60°F for a pressure drop of one psi. It can be shown from Darcy's equation that with C_v measured in this manner:

$$C_v = \frac{29.9\,d^2}{(fL/D)^{1/2}} \qquad (8-33)$$

where D = fitting equivalent ID, ft
d = fitting equivalent ID, in.
L = fitting equivalent length, ft

Derivation of Equation 8-33

Q_l is in bpd, C_v in gpm, ΔP in psi, d in inches, D in ft,

$$\Delta P = (11.5 \times 10^{-6})\frac{f\,L\,Q_l^2(S.G.)}{d^5}$$

$$\Delta P = 1$$

$$Q_l = C_v\,\frac{gal}{min} \times \frac{1}{42}\,\frac{bbl}{gal} \times 60\,\frac{min}{hr} \times 24\,\frac{hr}{day}$$

$$Q_l = 34.3\,C_v$$

$$S.G. = 1$$

$$d = 12D$$

$$(34.3\,C_v)^2 = \frac{1\,d^4}{11.5 \times 10^{-6}\,(fL/12D)}$$

$$C_v = \frac{29.9\,d^2}{(fL/D)^{1/2}}$$

It follows from the definition of K_r that the relationship between C_v and K_r is:

$$C_v = \frac{29.9\,d^2}{(K_r)^{1/2}} \qquad (8-34)$$

The pressure drop for any valve or fitting for which C_v is known is derived as follows:

$$\Delta P = 8.5 \times 10^{-4}\left[\frac{Q_l}{C_v}\right]^2 (S.G.) \qquad (8-35)$$

where Q_l = Liquid flow rate, BPD
S.G. = Liquid specific gravity relative to water

Derivation of Equation 8-35

Q_l is in BPD, ΔP in psi

$$\Delta P = (11.5 \times 10^{-6}) \frac{f\,L\,Q_l^2 (S.G.)}{d^5}$$

$$Q_l = C_v \frac{gal}{min} \times \frac{1}{42} \frac{bbl}{gal} \times 60 \frac{min}{hr} \times 24 \frac{hr}{day} = 34.3 C_v$$

$$\Delta P = 1 \text{ psi}, \quad S.G. = 1$$

$$1 = (11.5 \times 10^{-6}) \left[\frac{fL\,(34.3\,C_v)^2}{d^5} \right]$$

$$\frac{fL}{d^5} = \frac{73.9}{C_v^2}$$

$$C_v^2 = \frac{73.9\,d^4}{\left[\frac{fL}{d} \right]}$$

$$d = 12D$$

$$C_v = \frac{29.9\,d^2}{\left[\frac{fL}{D} \right]^{1/2}} = \frac{29.9\,d^2}{(K_r)^{1/2}}$$

$$\Delta P = (11.5 \times 10^{-6}) \frac{73.9}{C_v^2} Q_l^2\ (S.G.)$$

$$\Delta P = 8.5 \times 10^{-4} \left[\frac{Q_l}{C_v} \right]^2 (S.G.)$$

Equivalent Length

It is often simpler to treat valves and fittings in terms of their equivalent length of pipe. The equivalent length of a valve or fitting is the length of an equivalent section of pipe of the same diameter that gives the same head loss. Total head loss or pressure drop is determined by adding all equivalent lengths to the pipe length. The equivalent length, L_e, can be determined from K_r or C_v as follows:

$$L_e = \frac{K_r D}{f} \tag{8-36}$$

$$L_e = \frac{K_r d}{12f} \tag{8-37}$$

$$L_e = \frac{74.5d^5}{fC_v^2} \tag{8-38}$$

Table 8-4 summarizes the equivalent length for various commonly used valves and fittings. Figures 8-11 and 8-12 show equivalent lengths of various fabricated bends.

Laminar Flow Coefficient

Equivalent lengths that are usually published in tables are for turbulent flow. The following equation is used when the flow is laminar, that is, the Reynolds number is less than 1,000:

$$(L_e)_{laminar} = \frac{Re}{1{,}000} L_e \tag{8-39}$$

where $(L_e)_{laminar}$ = equivalent length to be used in pressure drop calculations (never less than actual fitting length)
L_e = equivalent length of the valve or fitting if flow were turbulent

EXAMPLE PRESSURE DROP CALCULATIONS

Example 8-1: Pressure Drop in Liquid Line

Given: Flow rates: Condensate = 800 bpd
Water = 230 bpd
Specific gravity: Condensate = 0.87
Water = 1.05
Viscosity = 3 cp
Length = 7,000 ft
Inlet pressure = 900 psi
Temperature = 80°F

Problem: Solve for pressure drop in a 2-inch and 4-inch I.D. line using the general equation and Hazen-Williams.

Table 8-4
Equivalent Length of Valves and Fittings in Feet

Nominal Pipe size in.	Globe valve or ball check valve	Angle valve	Swing check valve	Plug cock	Gate or ball valve	45° ell		Short rad. ell		Long rad. ell		Hard T.		Soft T.		90° miter bends			Enlargement					Contraction				
																			Sudden			Std. red.		Sudden			Std. red.	
																			Equiv. L in terms of small d									
						Weld	thrd	Weld	thrd	Weld	thrd	Weld	thrd	Weld	thrd	2 miter	3 miter	4 miter	d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4	d/D = 1/4	d/D = 1/2	d/D = 3/4	d/D = 1/2	d/D = 3/4
1½	55	26	13	7	1	1	2	3	5	2	3	8	9	2	3				5	3	1	4	1	3	2	1	1	
2	70	33	17	14	2	2	3	4	5	3	4	10	11	3	4				7	4	1	5	1	3	3	1	1	
2½	80	40	20	11	2	2	..	5	..	3	..	12		3	..				8	5	2	6	2	4	3	2	2	
3	100	50	25	17	2	2		6		4		14		4					10	6	2	8	2	5	4	2	2	
4	130	65	32	30	3	3		7		5		19		5					12	8	3	10	3	6	5	3	3	
6	200	100	48	70	4	4		11		8		28		8					18	12	4	14	4	9	7	4	4	1
8	260	125	64	120	6	6		15		9		37		9					25	16	5	19	5	12	9	5	5	2
10	330	160	80	170	7	7		18		12		47		12					31	20	7	24	7	15	12	6	6	2
12	400	190	95	170	9	9		22		14		55		14		28	21	20	37	24	8	28	8	18	14	7	7	2
14	450	210	105	80	10	10		26		16		62		16		32	24	22	42	26	9			20	16	8		
16	500	240	120	145	11	11		29		18		72		18		38	27	24	47	30	10			24	18	9		
18	550	280	140	160	12	12		33		20		82		20		42	30	28	53	35	11			26	20	10		
20	650	300	155	210	14	14		36		23		90		23		46	33	32	60	38	13			30	23	11		
22	688	335	170	225	15	15		40		25		100		25		52	36	34	65	42	14			32	25	12		
24	750	370	185	254	16	16		44		27		110		27		56	39	36	70	46	15			35	27	13		
30				312	21	21		55		40		140		40		70	51	44										
36					25	25		66		47		170		47		84	60	52										
42					30	30		77		55		200		55		98	69	64										
48					35	35		88		65		220		65		112	81	72										
54					40	40		99		70		250		70		126	90	80										
60					45	45		110		80		260		80		190	99	92										

(Courtesy of G.P.S.A. Engineering Data Book)

Solution to Example 8-1

1. *General Equation*

Specific gravity of liquid:

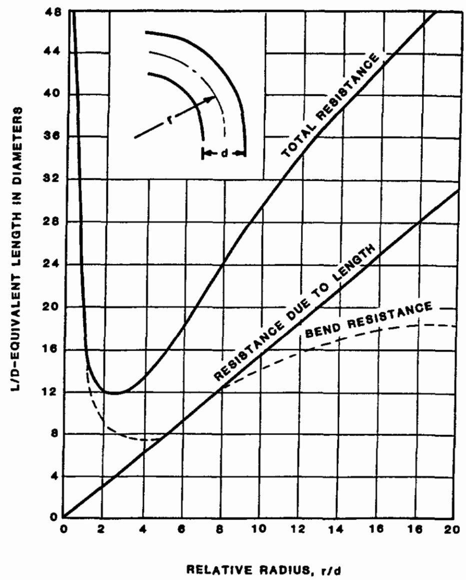


Figure 8-11. Equivalent length of 90° bends (*courtesy of Crane Technical Paper 410*).

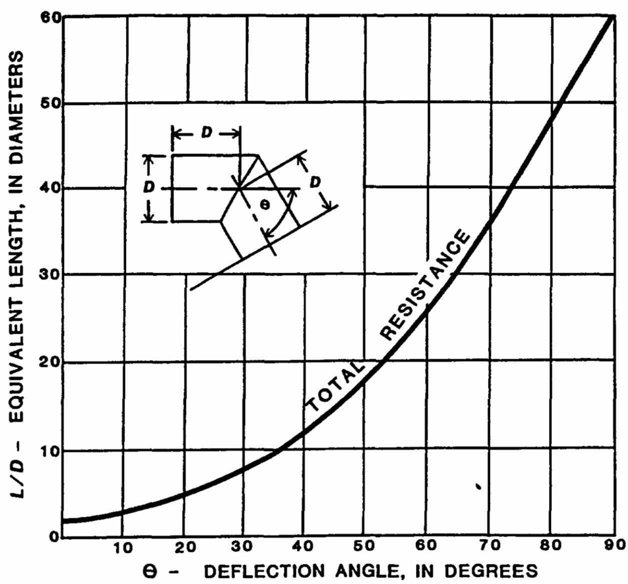


Figure 8-12. Equivalent length of mitre bends (*courtesy of Crane Technical Paper 410*).

$$S.G. = \frac{(230)(1.05)}{1{,}030} + \frac{(800)(0.87)}{1{,}030} = 0.91$$

$$Re = \frac{(92.1)(0.91)(1{,}030)}{3d} = \frac{28{,}775}{d}$$

$\varepsilon = 0.004$ (assume old steel)

$$\Delta P = \frac{(11.5 \times 10^{-6})(7{,}000)(1{,}030)^2(0.91)f}{d^5}$$

$$= \frac{77{,}725\,f}{d^5}$$

	Diameter 2-in.	4-in.
Re	14.4×10^3	7.2×10^3
ε/d	0.002	0.001
f (from chart)	0.032	0.034
ΔP	78 psi	2.6 psi

2. *Hazen-Williams*

$C = 120$ (assume)

$$H_L = 0.015\,\frac{(1{,}030)^{1.85}(7{,}000)}{(120)^{1.85}\,d^{4.87}}$$

$$H_L = \frac{5{,}604}{d^{4.87}}$$

$$\Delta P = \frac{(0.91)(62.4)}{144} H_L = 0.39\,H_L$$

Diameter = 2 in. $\Delta P = 74.7$ psi

Diameter = 4 in. $\Delta P = 2.6$ psi

Example 8-2: Pressure Drop in Gas Line

Given: Flow rates: Gas = 23 MMscfd
Viscosity = 3 cp
Gravity: Gas = 0.85
Length = 7,000 ft

Inlet pressure = 900 psi
Temperature = 80°F

Problem: Solve for pressure drop in a 4-in. and 6-in. ID line using the:

1. General equation
2. Assumption of $\Delta P < 10\% \; P_1$
3. Panhandle B Equation
4. Weymouth Equation.

Solution to Example 8-2

1. *General Equation*

Specific gravity of liquid:

Gas viscosity = 0.013 (from chart in Chapter 3)

$$Re = \frac{(20,100)(23)(0.85)}{(0.013)\,d} = \frac{30,227,000}{d}$$

$\varepsilon = 0.004$ (assume old steel)

$Z = 0.67$ (from chart in Chapter 3)

$$P_1^2 - P_2^2 = 25.2\,\frac{(0.85)(23)^2\,(0.67)(540)(7,000)f}{d^5} = \frac{2.87 \times 10^{10}(f)}{d^5}$$

	Diameter 4-in.	6-in.
Re	7.6×10^6	5.0×10^6
ε/d	0.001	0.0007
f (from chart)	0.0198	0.0180
$P_1^2 - P_2^2$	555×10^3	66×10^3
P_2	505	863
ΔP	395 psi	37 psi

2. *Approximate Equation*

$$\Delta P = \frac{(12.6)\;(0.85)(540)(0.67)(7,000)(23)^2 f}{(900)d^5} = \frac{1.59 \times 10^7 (f)}{d^5}$$

Diameter = 4 in.

$\Delta P = 308$ psi

Diameter = 6 in.

$\Delta P = 37$ psi

3. *Panhandle B Equation*

$$L_m = \frac{7{,}000}{5{,}280} = 1.33 \text{ miles}$$

$$E = 0.95 \text{ (assumed)}$$

$$23 = (0.028)(0.95)\left[\frac{(900)^2 - P_2^2}{(0.85)^{0.95}(0.67)(540)(1.33)}\right]^{0.51} d^{2.53}$$

$$\frac{810 \times 10^3}{412} - \frac{P_2^2}{412} = \left[\frac{23}{(0.028)(0.95)}\right]^{1.96}\left[\frac{1}{d}\right]^{4.96}$$

$$P_2^2 = 810 \times 10^3 - \frac{235 \times 10^6}{d^{4.96}}$$

	Diameter 4-in.	6-in.
P_2	753	882 psi
ΔP	147	18 psi

4. *Weymouth*

$$23 = 1.11\, d^{2.667}\left[\frac{(900)^2 - P_2^2}{(7{,}000)(0.85)(0.67)(540)}\right]^{1/2}$$

$$\frac{810 \times 10^3}{2.2 \times 10^6} - \frac{P_2^2}{2.2 \times 10^6} = \left[\frac{23}{1.11}\right]^2 \frac{1}{d^{5.33}}$$

$$P_2^2 = 810 \times 10^3 - \frac{9.44 \times 10^8}{d^{5.33}}$$

	Diameter 4-in.	6-in.
P_2	476	862 psi
ΔP	424 psi	38 psi

Example 8-3: Pressure Drop in Two-Phase Line

Given: Same conditions as Examples 8-1 and 8-2

Problem: Solve for the pressure drop with both liquid and gas flow in a single 4-in., 6-in., or 8-in. line.

Solution to Example 8-3

Specific gravity of liquid = 0.91 (from before)

Z = 0.67 (from chart in Chapter 3)

Flow rate:

$$W = (3{,}180)(23)(0.85) + (14.6)(1{,}030)(0.91)$$
$$= 75{,}854 \text{ lb/hr}$$

Gas liquid ratio:

$$R = \frac{23{,}000{,}000}{1{,}030} = 22{,}330 \text{ ft}^3/\text{bbl.}$$

Density of mixture (compute at 900 psi):

$$\rho_m = \frac{(12{,}409)(0.91)(915) + (2.7)(22{,}330)(0.85)(915)}{(198.7)(915) + (22{,}330)(540)(0.67)}$$

$$\rho_m = 6.93 \text{ lb/ft}^3$$

$$\Delta P = \frac{(6.9\times10^{-8})(7{,}000)(75{,}854)^2}{(6.93)\,d^5}$$

$$= \frac{401\times10^3}{d^5}$$

Line ID	ΔP
4 in.	392 psi
6 in.	52 psi
8 in.	12 psi

It may be desirable to iterate using an average density for each case.

CHAPTER

9

*Choosing a Line Size and Wall Thickness**

INTRODUCTION

Chapter 8 discussed the basic principles for determining pressure drop in piping. Flow equations for liquid flow, compressible flow, and two-phase flow, and how pressure drops in valves and fittings are calculated when using the various flow equations were discussed. This chapter discusses the concepts involved in choosing a line size and determining the wall thickness of the pipe.

The first part of this chapter introduces the basic concept of erosional flow and discusses velocity and pressure drop criteria used to determine the required inside diameter for liquid, gas, and two-phase flow piping. The second part discusses criteria for selecting the wall thickness required so that a pipe of a given diameter is capable of containing a specified pressure. The third part discusses pressure rating classes used for fittings and valves to withstand a specified pressure. The last part of

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*Reviewed for the 1998 edition by Robert S. Ferguson of Paragon Engineering Services, Inc.

this chapter includes some example calculations for choosing a line size and wall thickness.

LINE SIZE CRITERIA

When choosing a line size it is necessary to consider both pressure drop and velocity of flow. The line needs to be large enough so that the pressure available will drive the fluid through the line. Normally, pressure drop is not a governing criterion in production facility piping systems, since most of the pressure drop occurs across a control valve and there is relatively little pressure drop in the line compared to that available in the process.

Pressure drop for a given line diameter can be calculated using the techniques discussed in Chapter 8. Pressure drop is a particularly important criterion for long lines or those flowing between pieces of equipment operating at the same or nearly the same pressure. In calculating pressure drop, especially for flow between low pressure and atmospheric vessels, equivalent lengths and elevation changes must be considered.

Line diameter must also be sized for a minimum and a maximum velocity. The fluid must be kept below some maximum velocity to prevent such problems as erosion, noise, and water hammer. The fluid must also be kept above some minimum velocity to minimize surging and to transport sand and other solids.

Erosional Flow

Fluid erosion occurs when liquid droplets impact the wall with enough force to erode the products of corrosion, exposing the metal to the fluid and allowing more corrosion to occur. At even higher impact forces it is possible for the steel itself to be eroded. The higher the velocity of flow, the greater the tendency for fluid erosion to occur. Experiments in two-phase flow systems indicate that erosion of the products of corrosion occurs when the velocity of flow exceeds the value given by:

$$V_e = \frac{C}{(\rho_m)^{1/2}} \tag{9-1}$$

where V_e = erosional flow velocity, ft/sec
ρ_m = density of the fluid, lb/ft^3
C = empirical constant

Before 1990, API Recommended Practice 14E, "Design and Installation of Offshore Production Platform Piping Systems," suggested a value of C = 100 for continuous service and 125 for non-continuous service. Analysis of field data indicates that constants higher than 100 can be used if corrosion is controlled. In 1990, API RP 14E was rewritten; the newer edition suggests that values of C from 150 to 200 may be used for continuous, non-corrosive or corrosion controlled services, if no solids are present. For solids-free intermittent services, API RP 14E states that industry experience shows that values of C up to 250 have been used successfully. Where sand production is anticipated, fluid velocities should be reduced, and periodic surveys of the pipe wall thickness should be considered. Pipe designs should include the use of sand probes, a 3-foot minimum run of straight pipe downstream of choke outlets, long-radius ells, or target tees.

Although no universally accepted system for erosional velocity exists to date, conclusions can be drawn from a recent Southwest Research Institute (SwRI) project for API RP 14E [1]. SwRI suggests that the erosional velocity equation (9-1) is not appropriate for all possible flow regimes. SwRI recommends dividing the erosional velocity criteria into four different groups:

- Clean service—no solids or corrosion present
- Erosive service—solids present in the flow stream with no corrosion
- Corrosive service—corrosion present without solids
- Erosive and corrosive service—both solids and corrosive media present

Clean service test results conclude that erosional velocity limitations are not required. Velocities for this flow condition should be limited to 60 ft/sec to prevent excessive noise.

For erosive service, the erosional velocity can be determined from the following equation:

$$V_e = K_s \frac{d}{\sqrt{Q_s}} \tag{9-2}$$

where V_e = fluid erosional velocity, ft/sec
K_s = fitting factor from Table 9-1
d = pipe inside diameter, in
Q_s = solids (sand) flow rate, ft^3/day

Table 9-1 lists the appropriate values of K_s to be used in Equation 9-2. Experimental test results for corrosive service and erosive/corrosive service are not yet available.

Table 9-1
Erosive Service Fitting Factor Table

Fitting Type	Radius to Diameter Ratio	Material	K_s Factor: Dry Gas Flow	K_s Factor: Mist Flow	K_s Factor: Liquid Flow
Elbow	1.5	ASTM 216-WBC	0.95	0.84	44.51
		ASTM A234-WPB	1.49	1.34	
	2.0	ASTM 216-WBC	1.00	0.91	44.51
		ASTM A234-WPB	1.58	1.46	
	2.5	ASTM A216-WBC	1.08	1.00	44.51
		ASTM A234-WPB	1.69	1.60	
	3.0	ASTM 216-WBC	1.15	1.10	37.62
		ASTM A234-WPB		1.73	
	3.5	ASTM 216-WBC	1.28	1.23	16.14
		ASTM A234-WPB	1.95	1.90	
	4.0	ASTM 216-WBC	1.48	1.41	14.07
		ASTM A234-WPB	2.10	2.01	
	4.5	ASTM 216-WBC	1.68	1.60	14.07
		ASTM A234-WPB	2.23	2.12	
	5.0	ASTM 216-WBC	1.99	1.90	14.07
		ASTM A234-WPB	2.38	2.28	
Plugged Tee	—	ASTM 216-WBC	8.73	5.56	20.75
		ASTM A234-WPB	12.85	7.04	14.07
Vortice Elbow		ASTM 216-WBC	15.94		26.60

(Courtesy of Southwest Research Institute)

Erosion of the pipe material itself can occur if solids are present in the fluid. There is no minimum velocity at which this will occur. One equation proposed to evaluate the erosion of metal is:

$$\text{vol} = \frac{K\ W\ V^2 \beta}{g\ P} \tag{9-3}$$

where vol = volume of metal eroded
V = particle velocity
P = penetration hardness of the material
β = a value between 0.5 and 1.0 depending upon the impingement angle of the particle
K = erosive wear coefficient
W = total weight of impinging solids particles

The form of this equation indicates that there is no threshold velocity at which erosion starts. Rather, erosion occurs at even small velocities and the amount of erosion increase with the square of the velocity. It can be seen from Equation 9-3 that the velocity for a given erosion rate is a function of 1/W. Since the percent of solids impinging on any surface is inversely proportional to the density of the fluid, the erosional velocity

can be expected to be proportional to the fluid density. This is contrary to the form of Equation 9-1. Thus, it is *not* correct to use Equation 9-1 with a low "C" value when solids are present as suggested by API RP 14E.

The rate of erosion depends on both the concentration of solids in the flow stream and the way in which these particles impinge upon the wall. At an ell, one would expect centrifugal force to cause a high percentage of the particles to impinge on the wall in a concentrated area. It can be shown that with a solids concentration of 10 lb/month in the flow stream the velocity for a 10 mil/year erosion rate in an ell can be as low as 5 ft/sec. At higher concentrations the erosional velocity would be even lower. For this reason, where sand production is anticipated, it is usually recommended that right angle turns in the pipe be accomplished with very long radius fabricated bends or target tees.

Figure 9-1 shows a target tee, and Figure 9-2 shows the greater life that can be expected through the use of a target tee instead of a long-radius ell.

Liquid Lines

The maximum velocity used in sizing liquid lines is on the order of 15 ft/sec. Experience has shown that this limit is normally sufficient to mini-

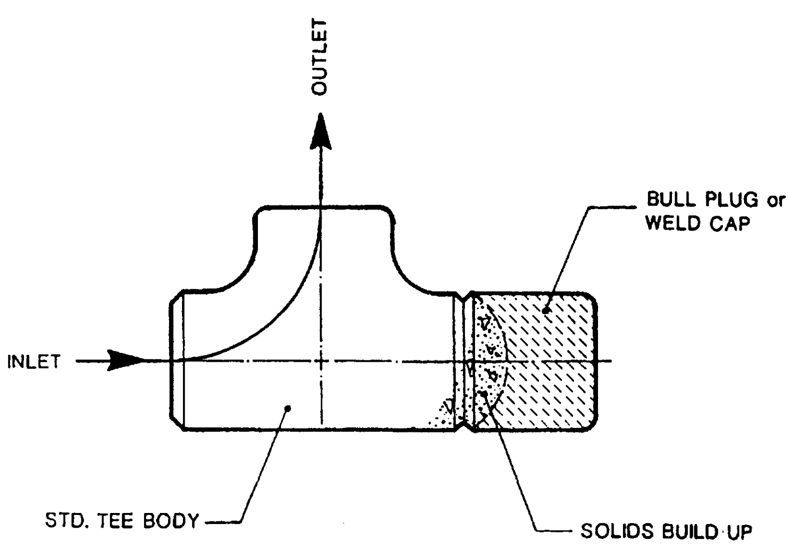
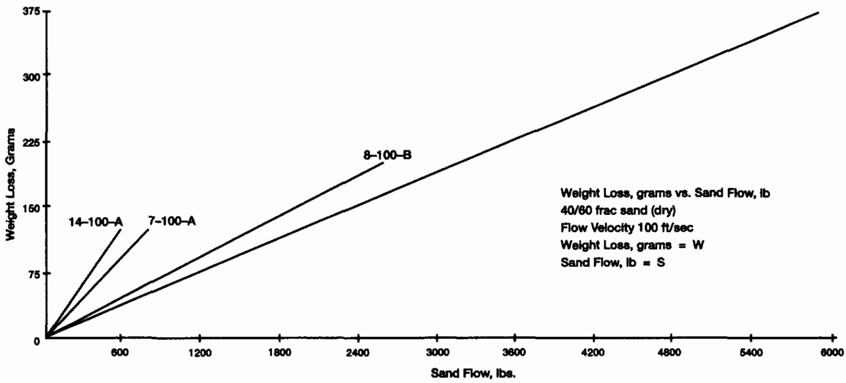


Figure 9-1. Target tee.



Run	Sand Flow, lb/min	Wear Rate	Fitting	Service Life, lb sand
14-100-A	3.11	W = .140 S	Water ell	785
7-100-A	3.04	W = .138 S	Field ell	800
8-100-B	3.00	W = .073 S	Bullplugged water tee	2600
8-100-A	2.95	W = .061 S	Bullplugged field tee	6000

Source: API OSAPR Project No. 2

Figure 9-2. Wear-rate comparison for standard fittings.

mize noise, water hammer, and erosion. Although Equation 9-1 was developed for two-phase flow, 15 ft/sec corresponds to a "C" value of 125 and is therefore consistent with the erosional criteria of Equation 9-1 in the limit as the gas flow rate approaches zero.

Liquid lines are normally sized to maintain a velocity sufficient to keep solid particles from depositing. If sand is transported in a pipe it will deposit on the bottom until a velocity of flow ("equilibrium velocity") builds up over the bed. At this point sand grains are being eroded from the bed at the same rate that they are being deposited. If the flow rate is increased the bed will be eroded until a new equilibrium velocity is reached and the bed is once again stabilized. If the flow rate is decreased, sand is deposited until a new equilibrium velocity is established and the bed stabilized. The equilibrium velocity follows a complex relationship and is reported in the literature [2]. In most practical cases a velocity of 3 to 4 ft/sec is sufficient to keep from building a sufficiently high bed to affect pressure drop calculations. For this reason a minimum velocity of 3 ft/sec is normally recommended.

Fluid velocity, expressed in oil field units, can be determined from the following equation:

$$V = 0.012 \frac{Q_l}{d^2} \qquad (9-4)$$

where V = fluid velocity, ft/sec
Q_l = liquid flow rate, bpd
d = pipe ID, in.

Derivation of Equation 9-4

Q is in ft^3/sec, A in ft^2, d in inches, V in ft/sec.

$$V = \frac{Q}{A}$$

$$A = \frac{\pi d^2}{(4)(144)}$$

where A = area, ft^2
d = pipe ID, in.
Q_l is in bpd

$$Q = Q_l \times \frac{5.61}{(24)(3,600)}$$

$$V = 0.012 \frac{Q_l}{d^2}$$

Equation 9-4 is expressed graphically in Figure 9-3 for different size pipe.

Gas Lines

As with liquid lines, one must make certain there is enough pressure available to move the gas through the pipe. Typically, this is a problem in either long gas gathering or transmission systems or in relief/vent piping where there is a very large instantaneous flow rate. Most other lines are so short that when they are sized for velocity considerations, pressure drop is not a problem.

For some lines the pressure lost due to friction must be recovered by recompressing the gas. When this is the case, it is possible to strike an economic balance between the cost of a larger pipe to minimize the pressure drop versus the cost of additional compression. Figure 9-4 is an approximate curve that attempts to strike this balance. In most cases, Figure 9-4 has little significance since the bulk of the pressure loss is due to a pressure control device, and the size and operating pressure of the compressor is not that closely controlled.

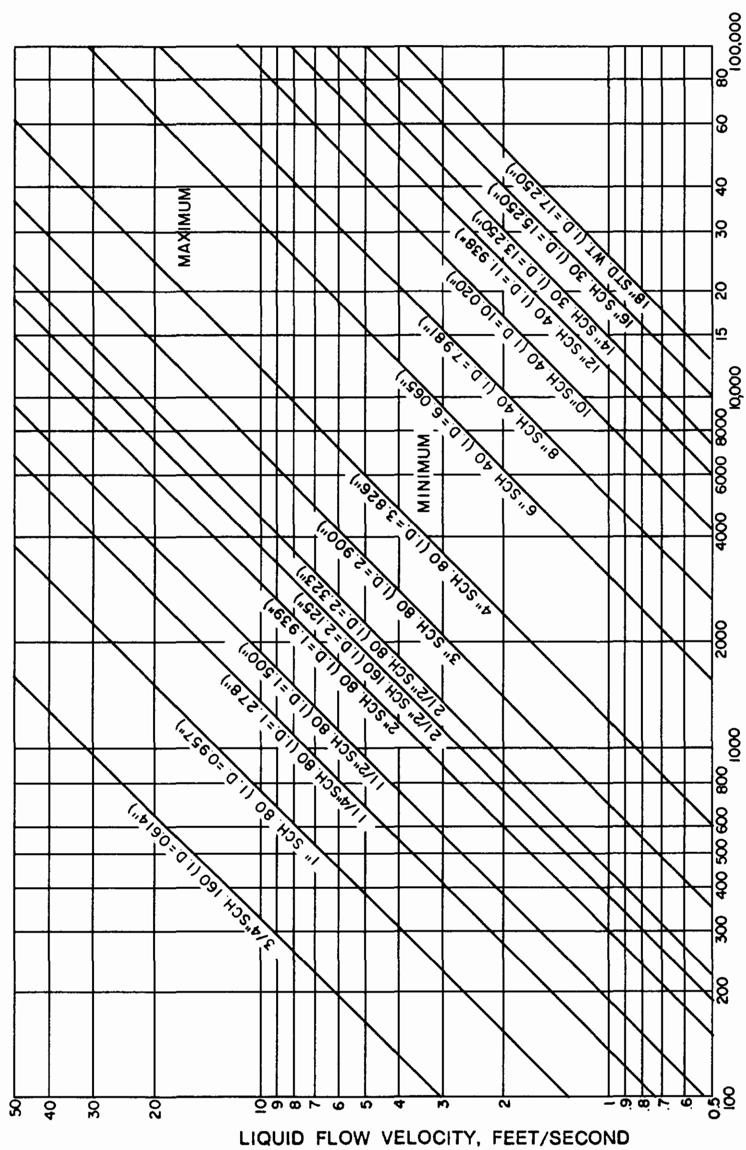


Figure 9-3. Velocity in liquid lines (courtesy of API RP 14E).

Figure 9-4 can be used to choose a pipe diameter directly, by rearranging the equation for pressure drop in a gas line with $\Delta P/P_1 < 10\%$ (Equation 8-17) for diameter. This is given by:

$$d^5 = \frac{1{,}260\, S\, Tf\, Q_g^2}{P\,(\Delta P/100\text{ ft})} \tag{9-5}$$

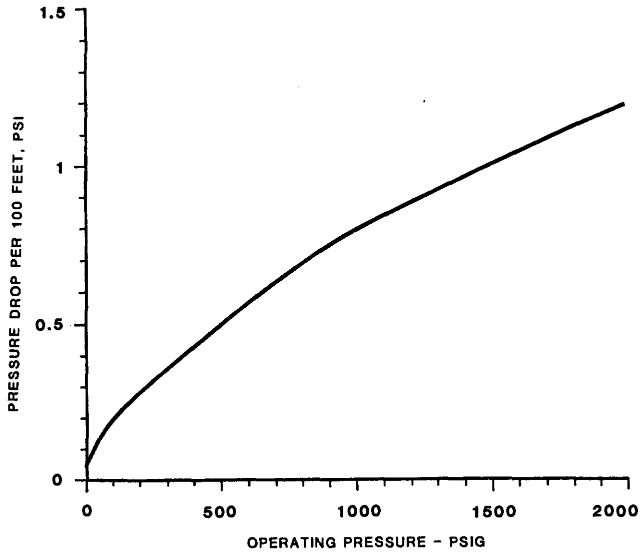


Figure 9-4. Acceptable pressure drop for short lines.

where d = pipe ID, in.
(S.G.) = specific gravity of gas (air = 1)
T = temperature, °R
f = Moody friction factor
Q_g = gas flow rate, MMscfd
P = pressure, psia
$\Delta P/100$ ft = desired pressure drop per 100 ft from Figure 9-4

As in liquid lines, gas lines must be kept between some maximum and minimum velocity. It is recommended that a minimum velocity of 10–15 ft/sec be maintained so as to minimize liquid settling out in low spots. Typically, gas velocities are normally kept below 60–80 ft/sec so as to minimize the effect of noise and corrosion.

Although the erosional criterion was derived for two-phase flow, we should check that in the limit as the liquid flow approaches zero this criterion is still met. Erosional velocity due to small amounts of liquid in the gas can be calculated from Equation 9-1 as:

$$V_e = 0.6\,C\left[\frac{T}{S\,P}\right]^{1/2} \qquad (9-6)$$

where V_e = erosional velocity, ft/sec
C = erosional flow constant

T = temperature, °R
S = specific gravity of gas at standard conditions (air = 1)
P = pressure, psia

Derivation of Equation 9-6

ρ_g is in lb/ft³

$$V_e = \frac{C}{(\rho_g)^{1/2}}$$

$$\rho_g = 0.0764\, S \times \frac{P}{14.7} \times \frac{520}{TZ}$$

$$\rho_g = 2.7\, \frac{S\,P}{TZ}$$

Assume $Z \cong 1.0$

$$V_e = 0.6\, C \left[\frac{T}{S\,P}\right]^{1/2}$$

For most instances with pressures less than 1,000 to 2,000 psi, the erosional velocity will be greater than 60 ft/sec and thus the erosional criteria will not govern. At high pressures, it may be necessary to check for erosional velocity before sizing lines for 60 ft/sec maximum velocity. In systems with CO_2 present in amounts as low as 1 to 2%, velocity should be limited to less than 50 ft/sec. Field experience has indicated that is it difficult to inhibit for CO_2 corrosion at higher velocities.

Actual gas velocity, expressed in oil field units, can be determined by

$$V = 60\, \frac{Q_g T Z}{d^2 P} \tag{9-7}$$

where Q_g = gas flow rate, MMscfd
T = temperature, °R
d = pipe ID, in.
P = pressure, psia
V = gas velocity, ft/sec
Z = gas compressibility factor

Derivation of Equation 9-7

V is in ft/sec, Q in ft^3/sec, A in ft^2, d in inches, Q_g in MMscfd.

$$V = \frac{Q}{A}$$

$$A = \frac{\pi d^2}{(4)(144)}$$

$$Q = Q_g \times \frac{1{,}000{,}000\ \text{scf}}{\text{MMscf}} \times \frac{\text{day}}{24\ \text{hr}} \times \frac{\text{hr}}{3{,}600\text{S}} \times \frac{14.7}{P} \times \frac{TZ}{520}$$

$$V = \frac{60\, Q_g\, TZ}{d^2\, P}$$

Two-Phase Flow

Typically, flowlines from wells, production manifolds, and two-phase gas/liquid pipelines are sized as two-phase lines. Gas outlets from separators or other process equipment contain some small amount of liquids in them but they are not considered two-phase lines. Similarly, liquid outlets from separators or other process equipment are usually considered single-phase liquid lines even though the pressure decreases across a liquid control valve or pressure loss in the line causes gas to evolve. The amount of gas evolved in liquid outlet lines will rarely be sufficient to affect a pressure loss calculation based on an assumption of liquid flow. A relatively large pressure drop is needed to evolve enough gas to affect this calculation. When a large pressure drop occurs in liquid system, maximum velocity is normally the governing criterion.

Since most two-phase lines operate at high pressure within the facility, pressure drop is usually not a governing criterion in selecting a diameter. However, pressure drop may have to be considered for some long flowlines from wells and in most two-phase gas/liquid pipelines.

It is recommended that a minimum flow velocity of 10–15 ft/sec be maintained to keep liquids moving in the line and thus minimize slugging of separator or other process equipment. This is very important in long lines with elevation changes. The maximum allowable velocity would be set by the minimum of 60 ft/sec for noise, 50 ft/sec if it is necessary to inhibit for CO_2 corrosion, or the erosional velocity. For two-phase flow the general erosional velocity equation, Equation 9-1, is usually expressed as:

$$V_e = \frac{C}{\rho_m^{1/2}} \tag{9-8}$$

where ρ_m = average density of the mixture at flowing pressure and temperature, lb/ft^3

The density of the mixture may be expressed by the following equation, which was derived in Chapter 8:

$$\rho_m = \frac{(12{,}409)\,(S.G.)\,P + (2.7)\,SRP}{(198.7)\,P + ZRT} \tag{9-9}$$

where S.G. = specific gravity of the liquid relative to water (use the average gravity for the hydrocarbon and water mixture)
R = gas/liquid ratio, ft^3/bbl
T = operating temperature, °R
S = specific gravity of the gas, at standard conditions (air = 1)

It can be shown that the minimum cross-sectional area of pipe for a maximum allowable velocity can be expressed as:

$$a = \left[\frac{9.35 + \dfrac{ZRT}{21.25P}}{1{,}000\ V}\right] Q_l \tag{9-10}$$

where a = minimum required cross-sectional area, in.^{2}
Q_l = liquid flow rate, bpd
V = maximum allowable velocity, ft/sec

This can be solved for pipe inside diameter:

$$d = \left[\frac{\left(11.9 + \dfrac{ZRT}{16.7P}\right) Q_l}{1{,}000\ V}\right]^{1/2} \tag{9-11}$$

Derivation of Equations 9-10 and 9-11

A is in ft^2, Q is in ft^3/sec, V is in ft/sec, R in ft^3/bbl

$$A = \frac{Q}{V}$$

From the derivation of Equation 8-30.,

$$Q = l + g = 6.49 \times 10^{-5}\, Q_l + 0.327 \frac{Q_g\, TZ}{P}$$

$$A = \frac{6.49 \times 10^{-5}\, Q_l + 0.327 \dfrac{Q_g\, TZ}{P}}{V}$$

$$R = \frac{1{,}000{,}000\, Q_g}{Q_l}$$

$$A = \frac{6.49 \times 10^{-5}\, Q_l + 0.327 \dfrac{RTZ\, Q_l}{10^6\, P}}{V}$$

a is in in.^{2}

$$A = \frac{a}{144}$$

Substitute and factor out Q_l and multiply top and bottom by 1,000

$$a = \left[\frac{9.35 + \dfrac{ZRT}{21.25P}}{1{,}000\ V}\right] Q_l$$

d is in inches

$$a = \frac{\pi\, d^2}{4}$$

$$d^2 = \left[\frac{11.9 + \dfrac{ZRT}{16.7\, P}}{1{,}000\ V}\right] Q_l$$

$$d = \left[\frac{\left(11.9 + \dfrac{ZRT}{16.7\, P}\, Q_l\right)}{1{,}000\ V}\right]^{1/2}$$

Figure 9-5 is a chart developed to minimize the calculation procedure. Care must be taken when utilizing this chart, as it is based on the

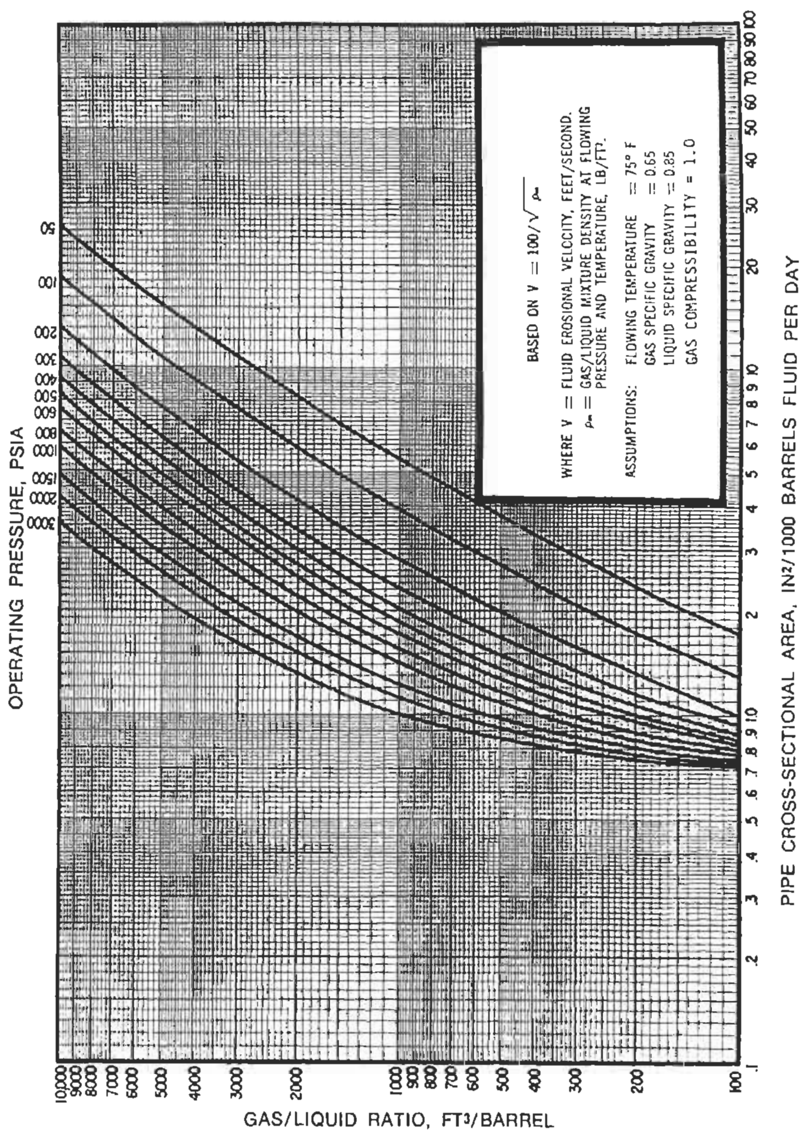


Figure 9-5. Erosional velocity chart (courtesy of *API RP 14E*).

assumptions listed. It is better to use Equations 9-8, 9-9, and 9-11 directly as follows:

1. Determine ρ_m from Equation 9-9.
2. Determine the erosional velocity, V_e from Equation 9-8.
3. For design use the smaller of V_e or that velocity required by the noise or CO_2 inhibition criteria.

4. Determine the minimum ID from Equation 9-11.
5. Check pressure drop, if applicable, to make certain there is enough driving force available.

WALL THICKNESS CRITERIA

After selecting the appropriate inside diameter, it is necessary to choose a pipe with enough wall thickness that it can withstand the internal pressure.

Standards and Requirements

There are different standards used throughout the world in calculating the required wall thickness of a pipe. The following is a list of the standards used in the United States. There are the ones most commonly used in oil production facility design and are similar to national standards that exist in other parts of the world.

1. *ANSI B 31.1—Power Piping.* This standard deals with steam and is required by the U.S. Coast Guard on all rigs.
2. *ANSI B 31.3—Chemical Plant and Petroleum Refinery Piping.* This standard is required by the U.S. Minerals Management Service for offshore platforms in federal waters. It is also used extensively for offshore facilities in state waters and for offshore facilities in other parts of the world.
3. *ANSI B 31.4— Liquid Petroleum Transportation Piping Systems.* This standard is normally used in onshore oil production facilities.
4. *ANSI B 31.8—Gas Transmission and Distribution Piping Systems.* This standard is normally used for gas lines in onshore production facilities and when transporting or distributing gas. In general, the U.S. Department of Transportation (DOT) has adopted this standard for gas pipelines, although it has modified some sections. The DOT regulations are in 49 CFR 192.

ANSI B 31.1 and ANSI B 31.3 use the same equation to calculate the required wall thickness. ANSI B 31.4 is actually a subset of ANSI B 31.8 when it comes to calculating wall thickness. Therefore, from a wall thickness standpoint, only ANSI B 31.3 and ANSI B 31.8 are in common use. In general, but not always, ANSI B 31.3 is the more severe in calculating required wall thickness.

General Hoop Stress Formula for Thin Wall Cylinders

Before discussing the determination of pipe wall thickness in accordance with the standards, it is necessary to introduce the concept of hoop stress. Figure 9-6 is a free body diagram of a length of pipe that was cut in half. The hoop stress in the pipe is considered a uniform stress over the thickness of the wall for a thin wall cylinder. Therefore, the force equilibrium equation can be expressed

$$2\sigma t L = P\,(d_o - 2t)L \qquad (9-12)$$

where σ = hoop stress in the pipe wall, psi
t = pipe wall thickness, in.
P = internal pipe pressure, psi
d_o = outside diameter of pipe, in.
L = pipe length, ft

Rearranging and solving for required wall thickness the equation reduces to

$$t = \frac{P\,d_o}{2(\sigma + P)} \qquad (9-13)$$

when only considering hoop stress. The standards build upon this formula.

ANSI B 31.3

The wall thickness specified by ANSI B 31.3 for a given pipe can be calculated by:

$$t = \left[t_c + t_{th} + \frac{P\,d_o}{2(SE + PY)}\right]\left[\frac{100}{100 - Tol}\right] \qquad (9-14)$$

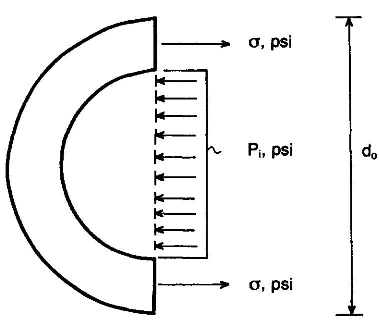


Figure 9-6. General hoop stress free body diagram.

where t = required wall thickness to be specified in ordering the pipe, in.

t_c = corrosion allowance, in. (normally 0.05 in.)

t_{th} = thread or groove depth, in. (Table 9-2)

P = internal pipe pressure, psi

d_o = pipe outside diameter, in.

S = allowable stress for pipe material, psi (Tables 9-3 and 9-4)

Table 9-2
Thread Allowance for Pipe Wall Thickness Calculations ANSI B 31.3

Nominal Pipe Size (in.)	t_{th}
¼–⅜	0.05
½–¾	0.06
1–2	0.08
2½–20	0.11

Table 9-3
Basic Allowable Stress for Grade B Seamless Pipe, psi

Temperature, °F	ASTM A106	API 5L
–20 to 100	20,000	20,000
200	20,000	19,100
300	20,000	18,150
400	20,000	17,250
500	18,900	16,350
600	17,300	15,550
650	17,000	15,000

Note: For additional information, see ANSI B 31.3, Appendix A

Table 9-4
Basic Allowable Stress for Other Grades of Pipe, ANSI B 31.3

Grade	Minimum Temperature (°F)	Allowable Stress Min. Temperature to 100°F
API 5L	–20	20,000
API 5LX-42	–20	20,000
API 5LX-46	–20	21,000
API 5LX-52	–20	24,000
ASTM A-106B	–20	20,000
ASTM A-333-6	–50	20,000
ASTM A-369-FPA	–20	16,000
ASTM A-524-FPB	–20	20,000
ASTM A-524-I	–20	20,000
ASTM A-524-II	–20	18,300

E = longitudinal weld joint factor
= 1.00 for seamless
= 0.85 for ERW
Y = factor
= 0.4 for ferrous materials below 900°F
Tol= manufacturers' allowed tolerance
= 12.5% for API 5L pipe up to 20-in. diameter
=10% for API 5L pipe greater than 20-in. diameter

In comparing Equations 9-13 and 9-14, it can be seen that the general hoop stress formula is contained in the code with the product of S times E, which can be considered an "allowable" stress, being set equal to the hoop stress. A factor, "Y," is added to account for deviation from the thin wall cylinder approximation used in deriving the hoop stress formula. The code requires that the wall thickness specified be sufficient so that, even in its assumed corroded condition where a thread or groove is cut, the hoop stress will not exceed the allowable. For this reason, t_c and t_{th} must be added to the hoop stress.

Manufacturers are allowed to furnish pipe with slightly less than the specified wall thickness as long as the pipe meets the tolerance requirements of the code under which the pipe is manufactured (normally API 5L). For this reason, ANSI B 31.3 makes the additional conservative assumption that one must specify a wall thickness so that at all locations in the pipe the stress is less than allowable. This is why the correction for tolerance appears in Equation 9-14.

ANSI B 31.3 allows for occasional variations above the allowable stress in accordance with the following criteria:

1. When the variation lasts no more than 10 hours at any one time and no more than 100 hours per year, it is permissible to exceed the pressure rating or the allowable stress for pressure design at the temperature of the increased condition by not more than 33%.

2. When the variation lists no more than 50 hours at any one time and not more than 500 hours per year, it is permissible to exceed the pressure rating or the allowable stress for pressure design at the temperature of the increased condition by not more than 20%.

For ease in picking a pipe wall thickness, tables such as Table 9-5 are published, giving the maximum allowable working pressure for standard pipe diameters and wall thicknesses as calculated from Equation 9-14 for a given grade of pipe.

Table 9-5
Maximum Allowable Working Pressures—Platform Piping
ASTM A106, Grade B, Seamless Pipe
(Stress Values from ANSI B 31.3—1980)
(Includes Corrosion/Mechanical Strength Allowance of 0.050 inches)

1	2	3	4	5	5	7	8	9	10
						MAXIMUM ALLOWABLE WORKING PRESSURES—PSIG			
Nominal Size In.	Outside Diameter In.	Nominal Wall Thickness In.	Nominal Weight Per Foot Lb	Weight Class	Schedule No.	-20/400°F	401/500°F	501/600°F	601/650°F
2	2.375	0.218	5.02	XS	80	2489	2352	2153	2115
		0.344	7.46	—	160	4618	4364	3994	3925
		0.436	9.03	XXS	—	6285	5939	5436	5342
2½	2.875	0.276	7.66	XS	80	2814	2660	2434	2392
		0.375	10.01	—	160	4194	3963	3628	3565
		0.552	13.70	XXS	—	6850	6473	5925	5822
		0.750	17.02	—	—	9772	9423	8625	8476
3	3.500	0.300	10.25	XS	80	2553	2412	2208	2170
		0.438	14.31	—	160	4123	3896	3566	3504
		0.600	18.58	XXS	—	6090	5755	5268	5176
4	4.500	0.237	10.79	STD	40	1439	1360	1245	1223
		0.337	14.98	XS	80	2276	2151	1969	1934
		0.438	18.98	—	120	3149	2976	2724	2676
		0.531	22.52	—	160	3979	3760	3442	3382
		0.674	27.54	XXS	—	5307	5015	4591	4511
6	6.625	0.280	18.97	STD	40	1206	1139	1043	1025
		0.432	28.57	XS	80	2062	1949	1784	1753
		0.562	36.42	—	120	2817	2663	2437	2395
		0.719	45.34	—	160	3760	3553	3252	3196
		0.864*	53.16	XXS	—	4660	4404	4031	3961
8	8.625	0.277	24.70	—	30	908	858	786	772
		0.322	28.55	STD	40	1098	1038	950	934
		0.406	35.66	—	60	1457	1377	1260	1238
		0.500	43.39	XS	80	1864	1762	1612	1584
		0.594	50.93	—	100	2278	2153	1970	1936
		0.719	60.69	—	120	2838	2682	2455	2413
		0.812*	67.79	—	140	3263	3084	2823	2774
		0.875*	72.42	XXS	—	3555	3359	3075	3022
		0.906*	74.71	—	160	3700	3496	3200	3145
10	10.750	0.250	28.04	—	20	636	601	550	541
		0.279	31.20	—	—	733	693	634	623
		0.307	34.24	—	30	827	781	715	703
		0.365	40.48	STD	40	1023	967	885	869
		0.500	54.74	XS	60	1485	1403	1284	1262
		0.594	64.40	—	80	1811	1712	1567	1540
		0.719	77.00	—	100	2252	2128	1948	1914
		0.844*	89.27	—	120	2700	2552	2336	2295
		1.000*	104.13	XXS	140	3271	3091	2829	2780
		1.125*	115.65	—	160	3737	3531	3232	3176

**All weld must be stress relieved.*

(Courtesy of API RP 14E)

Continued

Table 9-5
Continued

1	2	3	4	5	5	7	8	9	10
						MAXIMUM ALLOWABLE WORKING PRESSURES—PSIG			
Nominal Size In.	Outside Diameter In.	Nominal Wall Thickness In.	Nominal Weight Per Foot Lb	Weight Class	Schedule No.	-20/400°F	401/500°F	501/600°F	601/650°F
12	12.750	0.250	33.38	—	20	535	506	463	455
		0.330	43.77	—	30	760	719	658	646
		0.375	49.56	STD	—	888	839	768	755
		0.406	53.56	—	40	976	923	845	830
		0.500	65.42	XS	—	1246	1177	1078	1059
		0.562	73.22	—	60	1425	1347	1233	1212
		0.688	88.57	—	80	1794	1695	1552	1525
		0.844*	107.29	—	100	2258	2133	1953	1919
		1.000*	125.49	XXS	120	2730	2579	2361	2320
		1.125*	139.68	—	140	3114	2943	2694	2647
		1.312*	160.33	—	160	3700	3496	3200	3145
14	14.000	0.250	36.71	—	10	487	460	421	414
		0.312	45.68	—	20	645	610	558	549
		0.375	54.57	STD	30	807	763	698	686
		0.438	63.37	—	40	971	917	840	825
		0.500	72.09	XS	—	1132	1070	979	962
		0.594	85.01	—	60	1379	1303	1193	1172
		0.750	106.13	—	80	1794	1696	1552	1525
		0.938*	130.79	—	100	2304	2177	1993	1958
		1.094*	150.76	—	120	2734	2584	2365	2324
		1.250*	170.22	—	140	3171	2997	2743	2696
		1.406*	189.15	—	160	3616	3417	3128	3074
16	16.000	0.250	42.05	—	10	425	402	368	362
		0.312	52.36	—	20	564	533	488	479
		0.375	62.58	STD	30	705	666	610	599
		0.500	82.77	XS	40	988	934	855	840
		0.656	108.00	—	60	1345	1271	1164	1143
		0.843*	137.00	—	80	1780	1682	1540	1513
		1.031*	165.00	—	100	2225	2103	1925	1891
		1.218*	193.00	—	120	2675	2528	2314	2274
		1.437*	224.00	—	140	3212	3036	2779	2731
18	18.000	0.250	47.39	—	10	378	357	327	321
		0.312	59.03	—	20	501	473	433	425
		0.375	70.59	STD	—	626	591	541	532
		0.438	82.06	—	30	752	710	650	639
		0.500	93.45	XS	—	876	828	758	745
		0.562	105.00	—	40	1001	946	866	851
		0.718	133.00	—	60	1319	1246	1141	1121
		0.937*	171.00	—	80	1771	1674	1532	1506
		1.156*	208.00	—	100	2232	2109	1931	1897
		1.343*	239.00	—	120	2632	2487	2277	2237

** All welds must be stress relieved.*
(Courtesy of API RP 14E)

ANSI B 31.8

The wall thickness specified by ANSI B 31.8 and by 49 CFR 192 for a given pipe can be calculated by:

$$t = \frac{P\,d_o}{2(FETS)} \qquad (9-15)$$

where t = required wall thickness to be specified in ordering pipe, in.
P = internal pipe pressure, psi
d_o = pipe OD, in.
S = minimum yield strength of pipe, psi
F = design factor (Table 9-6)
E = longitudinal joint factor
= 1.0 for seamless, ERW, and flash weld
= 0.80 for furnace lap and electrical fusion welded pipe
= 0.60 for furnace butt welded pipe
T = temperature derating factor (Table 9-7)

Table 9-6
Design Factor, F

Location Class (B 31.8 Definition)	Class Location (DOT CFR 192)	Design Factor (F)	General Description*
Location Class 1, Division 1	Not Applicable	0.8	Sparsely populated areas, farmland, deserts
Location Class 1, Division 2	Class Location 1	0.72	Sparsely populated areas, farmland, deserts
Location Class 2	Class Location 2	0.6	Fringe areas around cities and towns
Location Class 3	Class Location 3	0.5	Residential and industrial areas
Location Class 4	Class Location 4	0.4	Dense areas with multi-story buildings

* *These descriptions are general in nature. A more specific description of locations for use of the different factors is included in ANSI B 31.8 and Department of Transportation requirements.*

Table 9-7
Temperature Derating Factor, T

Temperature, °F	Derating Factor
–20 to 250	1.000
300	0.967
350	0.933
400	0.900
450	0.867

In Equation 9-15, the term F times E times T times S represents an allowable stress. That is, the approximate safety factors for location class, joint type and temperature are applied to the yield strength of the pipe material to obtain an allowable stress. Equation 9-15 is merely a thin wall cylinder hoop stress equation with the assumption that $(\sigma + P) \cong \sigma$. This is a conservative assumption in that it overstates the wall thickness required.

Equations 9-14 and 9-15 are slightly different in form because of the different types of piping systems each was intended to cover. In Equation 9-15, no special provision is made for corrosion or thread allowance. Most gas distribution systems are large diameter, welded pipe. ANSI B 31.8 does not have a section on threaded or grooved joints as it assumes that all pipe is welded. If threaded pipe is used, consideration should be given to adding an allowance for thread or groove depth, as specified in ANSI B 31.3 (Table 9-2).

Most gas transmission lines handle a relatively "clean" product and so no specific wall thickness allowance is suggested for internal corrosion in ANSI B 31.8. In chemical plants and refineries a more corrosive product is normally handled. ANSI B 31.3 specifically states that an allowance should be included for corrosion and erosion. API Recommended Practice 14E for Offshore Production Platform Piping Systems suggests that a corrosion/mechanical strength allowance of 0.05 in. be used for carbon steel piping. This has become more or less a standard for production facility piping.

The main difference between Equations 9-14 and 9-15 is that Equation 9-14 uses a single allowable stress for a pipe material, whereas Equation 9-15 uses different design factors ("F") for different "location classes." That is, the assumption is made in ANSI B 31.3 that the cost of failure of a piping system is the same no matter where the chemical plant or refinery is located. On the other hand, ANSI B 31.8 recognizes that some gas transmission lines are located in sparsely settled areas where the cost of failure is low, others may be located in the middle of a suburban area where the potential for loss of life is greater, and still others may be located adjacent to large concentrations of people where the risk of loss of life is even greater. Thus, along the length of a gas transmission line, several different safety factors may be appropriate. This is considered by multiplying the pipe yield strength by a factor appropriate for a specific risk rather than specifying a single allowable stress for the material. Factors range from the most liberal (F = 0.8) to the most severe (F = 0.4).

The greater the consequence of failure, the lower the design factor, and thus the greater the required wall thickness.

Table 9-6 shows in general terms the location class and design factor, F, to use in different instances. A more specific description of locations for use of the different design factors comes from the ANSI code and Department of Transportation requirements. To determine the design factor it is necessary to first define a location class for the area in question. Unfortunately, the definition of location class is somewhat different between the ANSI Code and the Department of Transportation.

ANSI B 31.8 applies to gathering lines offshore and onshore, unless within a city or subdivision. The Code defines location class in terms of a "one-mile section."

To calculate the one-mile section layout, use a section one mile long and one-quarter mile wide along the pipeline route, with the pipeline on the center of the section. Count the dwellings intended for human occupancy. Each separate dwelling unit in a multiple dwelling is counted as a separate building intended for human occupancy.

Before 1989, ANSI B 31.8 designated the design factor by a construction type as either A, B, C or D. In 1989, the ANSI B 31.8 code was revised, and the term "construction type" was eliminated. The new designation is called the "location class." There are four classes of locations for ANSI B 31.8:

1. **Location Class 1**
   Location Class 1 refers to any one-mile section that has 10 or fewer buildings intended for human occupancy. This category includes areas such as wastelands, deserts, rugged mountains, grazing land, and farmland.

   Class 1 Division 1
   Location Class 1 Division 1 allows a maximum design factor (F) of 0.8 and requires a hydrostatic test pressure of 1.25 times the maximum operating pressure of the pipe.

   Class 1 Division 2
   Location Class 1 Division 2 allows a maximum design factor (F) of 0.72 and requires a hydrostatic test pressure of 1.1 times the maximum operating pressure of the pipe.

2. **Location Class 2**
   Location Class 2 refers to any one-mile section that has more than 10 but fewer than 46 buildings intended for human occupancy. This

category includes fringe areas around cities and towns, industrial areas, ranches, and country estates.

3. **Location Class 3**

   Location Class 3 refers to any one-mile section that has 46 or more buildings intended for human occupancy. This category includes suburban housing developments, shopping centers, residential areas, and industrial areas.

4. **Location Class 4**

   Location Class 4 includes locations where multistory buildings are prevalent, traffic is heavy or dense, and numerous utilities may be located underground.

The class location onshore for the Department of Transportation is the same as that for ANSI B 31.8, except that there is no Class 1 Division 1, and thus the highest design factor is 0.72.

There are several class boundaries. When a cluster of buildings intended for human occupancy requires a Class 2 or Class 3 location, the location ends 220 yards from the nearest building in the cluster. A Class 4 location ends 200 yards from the nearest building with four or more stories above ground.

The design factor for both ANSI B 31.8 and 49 CFR 192 is a function of the location class (or the class location). In populated areas, however, a different design factor may be required for pipelines located near roads, highways, and railroads. Table 9-8 shows the required design factors for both codes to be used for various locations.

Table 9-8
Design Factors for Steel Pipe Construction

Facility	Location Class 1, Division 1	Location Class 1, Division 2	Location Class 2	Location Class 3	Location Class 4
Pipelines, mains and service lines	0.8	0.72	0.6	0.5	0.4
Crossings of roads, railroads without casing:					
(a) Private roads	0.8	0.72	0.6	0.5	0.4
(b) Unimproved public roads	0.6	0.6	0.6	0.5	0.4
(c) Roads, highways, or public streets, with hard surface and railroads	0.6	0.6	0.5	0.5	0.4
Crossings of roads, railroads with casing:					
(a) Private roads	0.8	0.72	0.6	0.5	0.4
(b) Unimproved public roads	0.72	0.72	0.6	0.5	0.4
(c) Roads, highways, or public streets, with hard surface and railroads	0.72	0.72	0.6	0.5	0.4

Table 9-8
Continued

Facility	Location Class 1, Division 1	Location Class 1, Division 2	Location Class 2	Location Class 3	Location Class 4
Parallel encroachment of pipelines and mains on roads and railroads:					
(a) Private roads	0.8	0.72	0.6	0.5	0.4
(b) Unimproved public roads	0.8	0.72	0.6	0.5	0.4
(c) Roads, highways, or public streets, with hard surface and railroads	0.6	0.6	0.6	0.5	0.4
Fabricated assemblies	0.6	0.6	0.6	0.5	0.4
Pipelines on bridges	0.6	0.6	0.6	0.5	0.4
Compressor station piping	0.5	0.5	0.5	0.5	0.4
Near concentration of people in Location Class 1 and 2	0.5	0.5	0.5	0.5	0.4

(Courtesy of ANSI/ASME)

Table 9-9 lists allowable working pressures calculated from Equation 9-15 for various pipe grades, construction types, and normally available pipe diameters and wall thicknesses.

ANSI B 31.4

The required wall thickness equation for ANSI B 31.4 is the same as that for ANSI B 31.8 except the safety factor is fixed at F = 0.72 and there is no temperature derating factor. This is because the consequences of a leak in an oil line are not as severe as the consequences of a leak in a gas line. It is possible for a gas leak to lead quickly to an explosion and loss of life if a combustible cloud of gas comes in contact with a spark. An oil leak, on the other hand, provides a visual warning of its presence. It will typically spread more slowly to a source of combustion, giving ample warning to personnel in the vicinity to escape. While it may catch fire, it is unlikely to result in an explosion.

ANSI B 31.4 does not have a temperature derating factor ("T"), as it states that it is only applicable to temperatures between –20°F and 250°F.

Comparison of ANSI B 31.3 and B 31.8

ANSI B 31.8 requires a design factor of 0.5 for a compressor station. Figure 9-7 and 9-8 compare the wall thickness required for piping in a

(text continued on page 313)

Table 9-9
Gas transmission and distribution piping code for pressure piping ANSI B 31.8-1982
Carbon steel and high yield strength pipe
Values apply to A106, API 5L and API 5LX pipe having the same specified minimum yield strength as shown

NOM PIPE SIZE	O.D.	WALL THK.	ALLOWABLE WORKING PRESSURES UP TO 250°F, in PSIG — CONSTRUCTION TYPE DESIGN FACTORS																			
			TYPE A, F = 0.72*					TYPE B, F = 0.60					TYPE C, F = 0.50					TYPE D, F = 0.40				
			GR. B 35,000	42,000	46,000	52,000	60,000	GR.B 35,000	42,000	46,000	52,000	60,000	GR.B 35,000	42,000	46,000	52,000	60,000	GR.B 35,000	42,000	46,000	52,000	60,000
	(STD)	.154	3268					2723					2270					1816				
2	2.375	.218	4626					3855					3213					2570				
		.125	1800					1500					1250					1000				
		.156	2246					1872					1560					1248				
	3.500	.188	2707					2256					1880					1504				
3	(STD)	.216	3110					2592					2160					1728				
		.250	3600					3000					2500					2000				
		.281	4046					3372					2810					2248				
		.300	4320					3600					3000					2400				
		.125	1400	1680	1840			1167	1400	1533			973	1167	1278			778	933	1022		
		.156	1747	2097	2296			1456	1747	1913			1214	1456	1595			971	1165	1276		
		.188	2105	2526	2767			1754	2105	2306			1462	1755	1922			1170	1404	1537		
	4.500	.219	2453	2943	3223			2044	2453	2686			1704	2044	2239			1363	1635	1791		
4	(STD)	.237	2654	3185	3488			2212	2654	2907			1844	2212	2423			1475	1770	1938		
		.250	2800	3360	3680			2333	2800	3067			1945	2333	2556			1556	1869	2044		
		.281	3147	3776	4136			2623	3147	3447			2186	2622	2873			1748	2098	2298		
		.312	3494	4193	4593			2912	3494	3827			2427	2912	3190			1941	2330	2552		
		.337	3774	4530	4961			3145	3775	4134			2621	3146	3445			2097	2516	2756		
		.156	1187	1424	1560	1763		989	1187	1300	1469		824	989	1083	1224		659	791	866	980	
		.188	1429	1716	1880	2124		1192	1430	1567	1770		993	1192	1306	1475		794	954	1044	1180	
		.219	1666	2000	2190	2475		1388	1666	1825	2063		1157	1389	1521	1719		926	1111	1216	1375	
	6.625	.250	1902	2282	2500	2826		1585	1902	2083	2355		1321	1585	1736	1963		1057	1268	1389	1570	
6	(STD)	.280	2130	2556	2799	3164		1775	2130	2333	2637		1479	1775	1944	2198		1183	1420	1555	1758	
		.312	2373	2848	3120	3527		1978	2374	2600	2933		1649	1978	2167	2449		1319	1582	1733	1959	
		.375	2853	3424	3750	4237		2377	2853	3125	3531		1981	2378	2604	2943		1585	1902	2083	2354	
		.132	3287	3943	4319	4883		2739	3286	3599	4069		2283	2738	3000	3391		1826	2191	2400	2713	

Table 9-9
Continued

		.156	912	1094	1198	1354	760	912	998	1128	633	760	832	940	506	608	666	7512
		.188	1098	1318	1444	1632	915	1098	1203	1360	763	915	1003	1133	610	732	802	907
		.203	1186	1424	1559	1762	989	1186	1299	1469	824	989	1083	1224	659	791	866	979
		.219	1280	1535	1681	1901	1067	1280	1401	1584	889	1067	1168	1320	711	853	934	1056
		.250	1461	1753	1920	2170	1217	1461	1600	1809	1014	1217	1333	1507	812	974	1067	1206
		.277	1618	1942	2128	2405	1349	1618	1773	2004	1124	1349	1478	1670	899	1079	1182	1336
8	8.625	.312	1823	2189	2396	2709	1520	1823	1997	2258	1266	1520	1664	1881	1013	1216	1331	1505
	(STD)	.322	1882	2258	2473	2796	1568	1882	2061	2329	1307	1568	1717	1941	1045	1254	1374	1553
		.344	2011	2412	2642	2988	1676	2011	2202	2490	1396	1676	1835	2075	1117	1340	1468	1660
		.375	2191	2628	2880	3256	1826	2191	2399	2713	1521	1826	1999	2261	1217	1460	1599	1808
		.438	2560	3071	3364	3803	2133	2560	2804	3170	1778	2133	2336	2641	1422	1706	1869	2113
		.500	2922	3506	3840	4341	2435	2922	3200	3617	2029	2435	2667	3014	1623	1948	2133	2412
		.188	881	1058	1158	1310	733	881	965	1091	612	735	804	909	490	588	644	728
		.203	959	1143	1251	1415	794	952	1043	1179	661	794	869	983	529	635	695	786
		.219	1026	1231	1348	1525	855	1026	1124	1271	713	855	936	1059	570	684	749	847
		.250	1172	1407	1540	1741	977	1172	1284	1451	814	977	1070	1209	651	781	856	967
10	10.750	.279	1309	1570	1719	1944	1091	1309	1433	1620	909	1091	1194	1350	727	872	955	1080
		.307	1440	1728	1892	2138	1200	1440	1577	1782	1000	1200	1314	1486	800	960	1051	1189
		.344	1613	1935	2120	2396	1344	1613	1767	1997	1120	1344	1473	1664	896	1075	1178	1331
	(STD)	.365	1711	2054	2249	2542	1426	1711	1874	2119	1188	1426	1562	1766	951	1141	1249	1412
		.438	2054	2464	2700	3051	1712	2054	2250	2543	1426	1712	1875	2119	1141	1369	1500	1695
		.500	2344	2813	3081	3483	1953	2344	2567	2902	1628	1953	2140	2419	1302	1563	1712	1935
		.188	743	892	977	1104	619	743	814	920	516	619	678	767	413	495	543	613
		.203	803	963	1055	1193	689	803	879	995	558	669	733	829	446	535	586	663
		.219	866	1039	1138	1287	722	866	948	1073	601	722	790	894	481	577	632	715
		.250	988	1186	1299	1468	824	988	1082	1224	686	824	902	1020	549	659	722	816
		.281	1111	1332	1460	1651	926	1111	1217	1376	771	926	1014	1146	617	740	811	917
		.312	1233	1480	1620	1832	1028	1233	1350	1527	856	1028	1125	1273	685	822	900	1018
12	12.750	.330	1305	1566	1715	1939	1088	1305	1430	1616	906	1088	1191	1346	725	870	953	1077
		.344	1359	1631	1786	2020	1133	1359	1488	1683	944	1133	1240	1403	755	906	992	1122
	(STD)	.375	1482	1779	1948	2202	1235	1482	1624	1835	1029	1235	1353	1529	824	988	1082	1224
		.406	1606	1926	2110	2385	1338	1606	1758	1988	1115	1338	1465	1656	892	1070	1172	1325
		.438	1732	2077	2275	2572	1443	1732	1896	2144	1203	1443	1580	1786	962	1154	1264	1429
		.500	1976	2377	2598	2936	1647	1976	2165	2447	1373	1647	1804	2039	1098	1318	1443	1631

Type A construction also applicable to "Liquid Petroleum Transportation Piping Code," ANSI B 31.4-1971

(Courtesy GPSA Engineering Data Book)

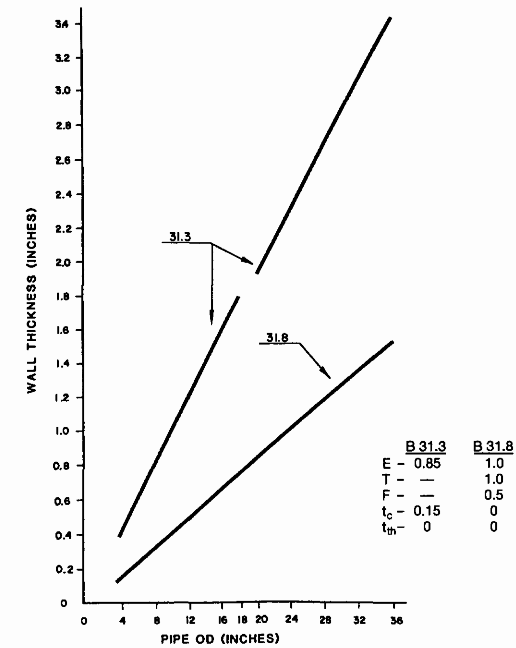


Figure 9-7. Wall thickness for 1,480 psi pressure API 5L-Grade B.

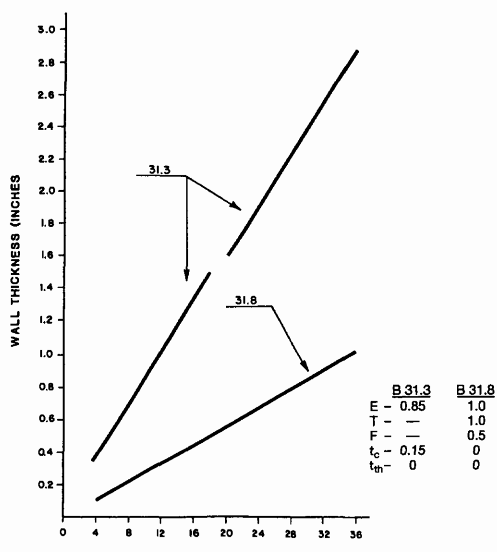


Figure 9-8. Wall thickness for 1,480 psi pressure API 5L-Grade X52.

(text continued from page 309)

compressor station depending upon which code is used. It can be seen that the B 31.3 code is more conservative than the B 31.8 code, especially when higher yield strength pipe material is used. This difference creates a problem in choosing the interface between a production facility designed to B 31.3 and a pipeline designed to B 31.8 or B 31.4. The location of the transition varies from company to company but it is usually at the plant fence for an onshore facility and at the first flange above the water on an offshore platform.

PRESSURE RATING CLASSES

Industry Standards

When designing piping systems one must consider piping components such as pipe flanges, fittings, and valves. These piping components must be able to withstand the stresses imposed by internal pressure. Unlike pipe, they are not straight cylinders but are of complex geometry and require a lot of study in order to determine the pressures they can withstand. Rather than requiring every designer to perform finite element analysis on each component, industry has developed standards on pipe flanges, fittings, and valves. The goal of the standards is to provide interchangeability between manufacturers, set dimensional standards, specify allowable service ratings for pressure and temperature ranges, specify material properties, and specify methods of production and quality control. The ANSI B 16.5 and API 6A specifications are the most commonly used. By specifying a specific pressure rating class that is rated for a pressure equaling or exceeding the maximum working pressure of the particular piping system, the designer is assured that all flanges, fittings, and valves furnished by any manufacturer will contain the pressure and have interchangeable dimensions.

The ANSI B 16.5 specification has seven classes of piping: 150, 300, 400, 600, 900, 1,500, and 2,500. Historically, the class designation was the allowable working pressure at 850°F. For example, the 300 ANSI class rating had a primary pressure rating of 300 psi at 850°F. The maximum non-shock pressure rating would increase as the temperature became cooler. With time, however, this distinction has become clouded and is no longer true.

Table 9-10 is a listing of the maximum non-shock working pressure rating for Material Group 1.1, as listed in the 1988 edition of ANSI B 16.5.

Table 9-10
Summary ANSI Pressure Ratings
Material Group 1.1

Class	–20 to 100	100 to 200
150	285	260
300	740	675
400	990	900
600	1,480	1,350
900	2,220	2,025
1,500	3,705	3,375
2,500	6,170	5,625

Material Group 1.1 includes most of the carbon steels commonly used in production facility piping. Table 9-10 lists temperature up to 200°F since most facility piping operates at temperatures less than 200°F. Table 9-11 contains additional information for other materials and for temperatures up to 1,500°F.

API 6A

The API 6A specification also has seven classes of piping: 2,000, 3,000, 5,000, 10,000, 15,000, 20,000, and 30,000. The API class designation is the maximum non-shock working pressure rating at 100°F. For example, 2,000 API class has a pressure rating of 2,000 psi at 100°F. Under certain conditions, API flange ratings up to 650°F are available. Consult API 6A for specific details.

API 6A requires more stringent control and testing of the metallurgy and methods of manufacture than does. ANSI B 16.5. As a result, even though 2,000, 3,000, and 5,000 API series flanges have the same dimensions and are completely interchangeable with 600, 900 and 1,500 ANSI series flanges, they have a higher pressure rating. Any joint made by bolting an API flange to its ANSI dimensional equivalent must be rated at the ANSI pressure rating. API 10,000, 15,000, 20,000, and 30,000 series have no ANSI series that are dimensionally equivalent.

Pipe, Valve, and Fitting Specifications

The purpose of pipe, valve, and fittings specifications is to determine, for a specific job, the governing industry codes, material requirements

(text continued on page 324)

Table 9-11A
American National Standard Pipe Flanges and Flanged Fittings (ANSI B16.5-1988)

Mat'l Group	Materials[4] (Spec-Grade)	See Notes	Mat'l Group	Materials[4] (Spec-Grade)	See Notes
1.1	A105, A181-II, A216-WCB, A515-70	(a)(h)	2.1	A182-F304, A182-F304H	—
	A516-70	(a)(g)		A240-304, A351-CF8	—
	A350-LF2, A537-C1.1	(d)		A351-CF3	(f)
1.2	A203-B, A203-E, A216-WCC	(a)(h)	2.2	A182-F316, A182-F316H, A240-316	—
	A350-LF3, A352-LC2, A352-LC3	(d)		A240-317, A351-CF8M	—
1.3	A352LCB	(a)		A351-CF3M	(g)
	A203-A, A203-D, A515-65	(a)	2.3	A182-F304L, A240-304L	(f)
	A516-65	(a)(j)		A182-F316L, A240-316L	(g)
1.4	A181-I, A515-60	(a)(h)	2.4	A182-F321, A240-321	(h)
	A516-60	(a)(g)		A182-F321H, A240-321H	—
	A350-LF1	(d)	2.5	A182-F347, A240-347	(h)
1.5	A182-F1, A204-A, A204-B, A217-WC1	(b)(h)		A182-F347H, A240-347H	—
	A352-LC1	(d)		A182-F348, A240-348	(h)
1.7	A204-C	(g)		A182-F348H, A240-F348H	—
	A182-F2, A217-WC4	(h)	2.6	A240-309S, A351-CH8, A351-CH20	—
	A217-WC5	(i)	2.7	A182-F310, A240-310S	(k)
1.9	A182-F11, A182-F12, A387-11, C1.2	(c)		A351-CK20	—
	A217-WC6	(j)			
1.10	A182-F22, A387-22, C1.2	(c)			
	A217-WC9	(j)			
1.13	A182-F5a, 217-C5	—			
1.14	A182-F9, A217-C12	—			

(table continued on next page)

Table 9-11A
Continued

Mat'l Group	Materials[4] (Spec-Grade)	See Notes	Mat'l Group	Materials[4] (Spec-Grade)	See Notes
3.1	B462, B463	(m)	3.7	B335-N10665, B333-N10665	(o)
	A351-CN7M	(o)	3.8	B574-N10276, B575-N10276	(o)
3.2	B160-N02200, B162	(m)		B574-N06455, B575-N06455	(o)
3.4	B564-N04400, B127	(m)		B446, B443	(m)
	B164-N04405	(m)		B335-N10001, B333-N10001	(m)
3.5	B564-N06600, B168	(m)		B573, B434, B572, B435	(m)
3.6	B564-N08800, B409-N08800	(m)			

NOTES:
1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. Temperature notes for all Material Groups, tables 2-150 through 2-5,000:
4. See table 1A for additional information and notes relating to specific materials.
(a) permissible but not recommended for prolonged use above about 800°F
(b) permissible but not recommended for prolonged use above about 850°F
(c) permissible but not recommended for prolonged use above about 1,100°F
(d) not to be used over 650°F
(f) not to be used over 800°F
(g) not to be used over 850°F
(h) not to be used over 1,000°F
(i) not to be used over 1,050°F
(j) not to be used over 1,100°F
(k) for service temperature 1,050°F and above, should be used only when assurance is provided that grain size is not finer than ASTM No. 6.
(m) use annealed material only
(o) use solution annealed material only

Table 9-11B
Class 150 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
−20 to 100	285	290	265	235	265		290				275	275	230	275	275	260		230	140	230	275	275	290		100
200	260	260	250	215			260				235	240	195	235	245	230		215	140	200	260	255	260		200
300	230	230	230	210			230				205	215	175	210	225	220		200	140	190	230	230	230		300
400						200					180	195	160	190		200			140	185		200			400
500						170					170		145			170			140			170			500
600						140					140		140			140			140			140			600
650						125					125		125			125						125			650
700						110					110		110			110						110			700
750						95					95		95			95						95			750
800						80					80		80			80						80			800
850						65					65		65			65									850
900						50					50					50									900
950						35					35					35									950
1000						20					20					20									1000

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150 through 2-2500.

Table 9-11C
Class 300 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp. °F	Carbon Steel				C	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
−20 to 100	740	750	695	620		750	750	750	750		720	720	600	720	720	670		600	360	600	720	720	750		100
200	675	750	655	560		750	710	715	750		600	620	505	610	635	605		555	360	530	670	660	750		200
300	655	730	640	550		730	675	675	730		530	560	455	545	590	570		525	360	495	640	625	730		300
400	635	705	620	530	640	705	660	650	705		470	515	415	495	555	535				480	615	600	705		400
500	600	665	585	500	620	665	640		665		435	480	380	460	520	505				480	595	580	665		500
600	550	605	535	455			605				415	450	360	435	490	480				480	575	575	605		600
650	535	590	525	450			590				410	445	350	430	480	465				480	565	570	590		650
700	535	570	520	450			570				405	430	345	420	470	455				480	555	565	570		700
750	505	505	475	445			530				400	425	335	415	460	445				470	530	530	530		750
800	410	410	390	370			510		500	510	395	415	330	415	455	435				460	510	505	510		800
850		270					485		440	485	390	405	320	410	445	425									850
900		170					450		355	450	385	395		405	430	415									900
950		105			280	345	380		260	370	375	385		385	385	385									950
1000		50			165	215	225	270	190	290	325	365		355	365	335	350								1000
1050						190	140	200	140	190	310	360		345	360	290	335								1050
1100							95	115	105	115	260	325		300	325	225	290								1100
1150							50	105	70	75	195	275		235	275	170	245								1150
1200							35	55	45	50	155	205		180	170	130	205								1200
1250											110	180		140	125	100	160								1250
1300											85	140		105	95	80	120								1300
1350											60	105		80	70	60	80								1350
1400											50	75		60	50	45	55								1400
1450											35	60		50	40	30	40								1450
1500											25	40		40	35	25	25								1500

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150.

Table 9-11D
Class 400 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp. °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nic-kel Al-loys	Temper-ature °F
−20 to 100	990	1000	925	825	925	1000	1000	1000	1000		960	960	800	960	960	895		800	480	800	960	960	1000		100
200	900	1000	875	750	905	1000	950	955	1000		800	825	675	815	850	805		740	480	705	895	885	1000		200
300	875	970	850	730	870	970	895	905	970		705	745	605	725	785	760		700	480	660	850	830	970		300
400	845	940	825	705	855	940	880	865	940		630	685	550	660	740	710			480	635	820	800	940		400
500	800	885	775	665	830	885	855		885		585	635	510	610	690	670			480	635	790	770	885		500
600	730	805	710	610			805				555	600	480	585	655	635			480	635	765	765	805		600
650	715	785	695	600			785				545	590	470	570	640	620				635	750	760	785		650
700	710	755	690	600			755				540	575	460	560	625	610				635	740	750	755		700
750	670	670	630	590			710				530	565	450	555	615	595				625		710			750
800	550	550	520	495			675		665	675	525	555	440	550	610	580				610		675			800
850		355					650		585	650	520	540	430	545	590	565									850
900		230					600		470	600	510	525		540	575	555									900
950		140			375	460	505		350	495	500	515		515	515	515									950
1000		70			220	285	300	355	255	390	430	485		475	485	450	465								1000
1050						250	185	265	190	250	410	480		460	480	390	445								1050
1100							130	150	140	150	345	430		400	430	300	390								1100
1150							70	140	90	100	260	365		315	365	230	330								1150
1200							45	75	60	70	205	275		240	230	175	275								1200
1250											145	245		185	165	135	215								1250
1300											110	185		140	125	105	160								1300
1350											85	140		110	90	80	105								1350
1400											65	100		80	70	60	75								1400
1450											45	80		65	55	40	50								1450
1500											30	55		50	45	30	30								1500

* Do not use ASTM A181 Grade I or II material.

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups.

Table 9-11E
Class 600 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp. °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
−20 to 100	1480	1500	1390	1235	1390	1500	1500	1500	1500		1440	1440	1200	1440	1440	1345		1200	720	1200	1440	1440	1500		100
200	1350	1500	1315	1125	1360	1500	1425	1430	1500		1200	1240	1015	1220	1270	1210		1115	720	1055	1345	1325	1500		200
300	1315	1455	1275	1095	1305	1455	1345	1355	1455		1055	1120	910	1090	1175	1140		1045	720	990	1275	1250	1455		300
400	1270	1410	1235	1060	1280	1410	1315	1295	1410		940	1030	825	990	1110	1065			720	955	1230	1200	1410		400
500	1200	1330	1165	995	1245	1330	1285	1280	1330		875	955	765	915	1035	1010			720	950	1185	1155	1330		500
600	1095	1210	1065	915			1210				830	905	720	875	985	955			720	950	1145	1145	1210		600
650	1075	1175	1045	895			1175				815	890	700	855	960	930				950	1130	1140	1175		650
700	1065	1135	1035	895			1135				805	865	685	840	935	910				950	1115	1130	1135		700
750	1010	1010	945	885			1065				795	845	670	830	920	895				935	1065				750
800	825	825	780	740			1015		995	1015	790	830	660	825	910	870				915	1015				800
850		535					975		880	975	780	810	645	815	890	850									850
900		345					900		705	900	770	790		810	865	830									900
950		205			560	685	755		520	740	750	775		775	775	775									950
1000		105			330	425	445	535	385	585	645	725		715	725	670	700								1000
1050						380	275	400	280	380	620	720		695	720	585	665								1050
1100							190	225	205	225	515	645		605	645	445	585								1100
1150							105	205	140	150	390	550		475	550	345	495								1150
1200							70	110	90	105	310	410		365	345	260	410								1200
1250											220	365		280	245	200	325								1250
1300											165	275		210	185	160	240								1300
1350											125	205		165	135	115	160								1350
1400											90	150		125	105	90	110								1400
1450											70	115		95	80	60	75								1450
1500											50	85		75	70	50	50								1500

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150.

Table 9-11F
Class 900 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp. °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
–20 to 100	2220	2250	2085	1850	2085	2250	2250	2250	2250		2160	2160	1820	2160	2160	2015		1800	1080	1800	2160	2160	2250		100
200	2025	2250	1970	1685	2035	2250	2135	2150	2250		1800	1860	1520	1830	1910	1815		1670	1080	1585	2015	1990	2250		200
300	1970	2185	1915	1640	1955	2185	2020	2030	2185		1585	1680	1360	1635	1765	1705		1570	1080	1485	1915	1870	2185		300
400	1900	2115	1850	1585	1920	2115	1975	1945	2115		1410	1540	1240	1485	1665	1600			1080	1435	1845	1800	2115		400
500	1795	1995	1745	1495	1865	1995	1925	1920	1995		1310	1435	1145	1375	1555	1510			1080	1435	1780	1735	1995		500
600	1640	1815	1600	1370	1815						1245	1355	1080	1310	1475	1435			1080	1435	1720	1720	1815		600
650	1610	1765	1570	1345	1765						1225	1330	1050	1280	1440	1395				1435	1690	1705	1765		650
700	1600	1705	1555	1345	1705						1210	1295	1030	1260	1405	1370				1435	1670	1690	1705		700
750	1510	1510	1420	1325	1595						1195	1270	1010	1245	1385	1340				1405	1595				750
800	1235	1235	1175	1110	1525				1490	1525	1180	1245	985	1240	1370	1305				1375	1520				800
850	805				1460				1315	1460	1165	1215	965	1225	1330	1275									850
900	515				1350				1060	1350	1150	1180		1215	1295	1245									900
950	310				845	1030	1130		780	1110	1125	1160		1160	1160	1160									950
1000	155				495	640	670	805	575	875	965	1090		1070	1090	1010	1050								1000
1050						565	410	595	420	565	925	1080		1040	1080	875	1000								1050
1100							290	340	310	340	770	965		905	965	670	875								1100
1150							155	310	205	225	585	825		710	825	515	740								1150
1200							105	165	135	155	465	620		545	515	390	620								1200
1250											330	545		420	370	300	485								1250
1300											245	410		320	280	235	360								1300
1350											185	310		245	205	175	235								1350
1400											145	225		185	155	135	165								1400
1450											105	175		145	125	95	115								1450
1500											70	125		115	105	70	70								1500

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150.

Table 9-11G
Class 1500 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2} gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials / Temp. °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316S	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Mo Cu Cb 20Cb	Nickel Alloy 200	Mo Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
−20 to 100	3705	3750	3470	3085	3470	3750	3750	3750	3750		3600	3600	3000	3600	3600	3360		3000	1800	3000	3600	3600	3750		100
200	3375	3750	3280	2810	3395	3750	3560	3580	3750		3000	3095	2530	3050	3180	3025		2785	1800	2640	3360	3310	3750		200
300	3280	3640	3190	2735	3260	3640	3365	3385	3640		2640	2795	2270	2725	2940	2845		2615	1800	2470	3190	3120	3640		300
400	3170	3530	3085	2645	3200	3530	3290	3240	3530		2350	2570	2065	2470	2770	2665			1800	2390	3070	3000	3530		400
500	2995	3325	2910	2490	3105	3325	3210	3200	3325		2185	2390	1910	2290	2590	2520			1800	2375	2965	2890	3325		500
600	2735	3025	2665	2285			3025				2075	2255	1800	2185	2460	2390			1800	2375	2870	2870	3025		600
650	2685	2940	2615	2245			2940				2040	2220	1750	2135	2400	2330				2375	2820	2845	2940		650
700	2665	2840	2590	2245			2840				2015	2160	1715	2100	2340	2280				2375	2785	2820	2840		700
750	2520	2520	2365	2210			2660				1990	2110	1680	2075	2305	2230				2340	2660	2650	2660		750
800	2060	2060	1955	1850			2540		2485	2540	1970	2075	1645	2065	2280	2170				2290	2540	2535	2540		800
850	1340						2435		2195	2435	1945	2030	1610	2040	2220	2125									850
900	860						2245		1765	2245	1920	1970		2030	2160	2075									900
950	515				1405	1715	1885		1305	1850	1870	1930		1930	1930	1930									950
1000	260				825	1065	1115	1340	960	1460	1610	1820		1785	1820	1680	1750								1000
1050						945	685	995	705	945	1545	1800		1730	1800	1460	1665								1050
1100							480	565	515	565	1285	1610		1510	1610	1115	1460								1100
1150							260	515	345	380	980	1370		1185	1370	860	1235								1150
1200							170	275	225	260	770	1030		910	855	650	1030								1200
1250											550	910		705	615	495	805								1250
1300											410	685		530	465	395	600								1300
1350											310	515		410	345	290	395								1350
1400											240	380		310	255	225	275								1400
1450											170	290		240	205	155	190								1450
1500											120	205		190	170	120	120								1500

* Do not use ASTM A181 Grade I or II material.

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150.

Table 9-11H
Class 2500 Pressure-Temperature Ratings
Pressures are in lbs/in.^{2}, gauge (psig)

Mat'l Group	1.1	1.2	1.3	1.4	1.5	1.7	1.9	1.10	1.13	1.14	2.1	2.2	2.3	2.4	2.5	2.6	2.7	3.1	3.2	3.4	3.5	3.6	3.7	3.8	
Materials Temp. °F	Carbon Steel				C ½Mo	½Cr-½Mo Ni-Cr-Mo	1 1¼Cr-½Mo	2¼Cr-1Mo	5Cr-½Mo	9Cr-1Mo	Type 304	Type 316S	Type 304L Type 316L	Type 321	Types 347 348	Type 309	Type 310	Cr Fe Ma Cu Cb 20Cb	Nickel Alloy 200	Ni Cu Alloys 400 405	Ni Cr Fe Alloy 600	Ni Fe Cr Alloy 800	Ni Mo Alloy B2	Nickel Alloys	Temperature °F
−20 to 100	6170	6250	5785	5145	5785	6250	6250	6250	6250		6000	6000	5000	6000	6000	5600		5000	3000	5000	6000	6000	6250		100
200	5625	6250	5470	4680	5660	6250	5930	5965	6250		5000	5160	4220	5080	5300	5040		4640	3000	4400	5600	5520	6250		200
300	5470	6070	5315	4560	5435	6070	5605	5640	6070		4400	4660	3780	4540	4900	4740		4360	3000	4120	5320	5200	6070		300
400	5280	5880	5145	4405	5330	5880	5485	5400	5880		3920	4280	3440	4120	4620	4440			3000	3980	5120	5000	5880		400
500	4990	5540	4850	4150	5180	5540	5350	5330	5540		3640	3980	3180	3820	4320	4200			3000	3960	4940	4820	5540		500
600	4560	5040	4440	3805	5040						3460	3760	3000	3640	4100	3980			3000	3960	4780	4780	5040		600
650	4475	4905	4355	3740	4905						3400	3700	2920	3560	4000	3880				3960	4700	4740	4905		650
700	4440	4730	4320	3740	4730						3360	3600	2860	3500	3900	3800				3960	4640	4700	4730		700
750	4200	4200	3945	3685	4430						3320	3520	2800	3460	3840	3720				3000	4430				750
800	3430	3430	3260	3085	4230				4145	4230	3280	3460	2740	3440	3800	3620				3820	4230				800
850	2230				4060				3660	4060	3240	3380	2680	3400	3700	3540									850
900	1430				3745				2945	3745	3200	3280		3380	3600	3460									900
950	860				2345	2860	3145		2170	3085	3120	3220		3220	3220	3220									950
1000	430				1370	1770	1860	2230	1600	2430	2685	3030		2970	3030	2800	2915								1000
1050						1570	1145	1660	1170	1570	2570	3000		2885	3000	2430	2770								1050
1100							800	945	860	945	2145	2685		2515	2685	1860	2430								1100
1150							430	860	570	630	1630	2285		1970	2285	1430	2060								1150
1200							285	460	370	430	1285	1715		1515	1430	1085	1715								1200
1250											915	1515		1170	1030	830	1345								1250
1300											685	1145		885	770	660	1000								1300
1350											515	860		685	570	485	660								1350
1400											400	630		515	430	370	460								1400
1450											285	485		400	345	260	315								1450
1500											200	345		315	285	200	200								1500

* Do not use ASTM A181 Grade I or II material.

NOTES:

1. Ratings shown apply to other material groups where column dividing lines have been omitted.
2. Provisions of Section 2 apply to all ratings.
3. See Temperature Notes for all Material Groups, Tables 2-150.

(text continued from page 314)

for pipe, flanges, fittings, bolts, nuts, and gaskets, material and construction for each valve used in the piping, welding certification and inspection requirements, and design details, (e.g., branch connections, pipe support spacing, clearances, and accessibility). Each pipe class in the job is assigned a designation that indicates its pressure rating and service. A separate table is prepared for each designation that lists the details of pipe material, end connection, acceptable valves of each type, fittings, etc. for the specific pipe designation. Often, each valve type is assigned a symbol or designation number that is referenced on the mechanical flow sheets. Tables 9-12 and 9-13 are examples of pipe, valve, and fittings tables.

A pipe, valve, and fitting specification differs from the industry standard pressure rating standards in that it addresses the quality of the piping components an construction details. It may require greater wall thicknesses than may be calculated from either ANSI B 31.3 or B 31.8. This is especially true for small diameter pipes where mechanical integrity can be more important than ability to withstand internal pressure. It designates the particular type of joint (e.g., flange, socket weld, screwed), that is required for each service and the quality of the valves and fittings to be purchased.

A discussion of each of the details in developing a pipe, valve, and fitting specification is beyond the scope of this chapter. A general description of items to consider is contained in API RP 14E.

EXAMPLES

Example No. 9-1: Liquid Line

Given: Same as Example 8-1.
Liquid flow to a low pressure separator operating at 150 psi.
Line is rated for 1,480 psi.

Problem: Choose a line size and wall thickness using B 31.3, B 31.4, and B 31.8.

(text continued on page 328)

Table 9-12
Example Pipe, Valves, and Fittings Table

150 LB ANSI
Non-Corrosive Service[1]

Temperature Range..-20 to 650°F
Maximum Pressure..Depends on flange rating[2] at service tempertaure

Size Ranges	General Specifications	Platform Service
Pipe	Grade Depends on Service	ASTM A106, Grade B, Seamless[3]
¾″ and smaller nipples	threaded and coupled	Schedule 160 or XXH
1½″ and smaller pipe	threaded and coupled	Schedule 80 min
2″ through 3″ pipe	beveled end	Schedule 80 min
4″ and larger pipe	beveled end	See Table 2.4
Valves (Do not use for temperatures above maximum indicated.)		
Ball		
½″ and smaller	1500 lb CWP AISI 316 SS screwed, regular port, wrench operated, Teflon seat	Manufacturers Figure No.____ (300°F)
¾″ through 1½″	1500 lb CWP, CS, screwed, regular port, wrench operated, Teflon seat	Manufacturers Figure No.____ or Figure No.____ (450°F)
2″ through 8″	150 lb ANSI CS RF flanged, regular port, lever or hand wheel operated, trunnion mounted	Etc.
10″ and larger	150 lb ANSI CS RF flanged, regular port, gear operated, trunnion mounted	Etc.
Gate		
½″ and smaller	2000 lb GWP, screwed, bolted bonnet, AISI 316 SS	Etc.
¾″ through 1½″	2000 lb CWP, screwed, bolted bonnet, forged steel	Etc.
2″ through 12″	150 lb ANSI CS RF flanged, standard trim, handwheel or lever operated	Etc.
Globe		
1½″ and smaller (Hydrocarbons)	2000 lb CWP CS screwed	Etc.
1½″ and smaller (Glycol)	2000 lb CWP CS socketweld	Etc.
2″ and larger	150 lb ANSI CS RF flanged, handwheel operated	Etc.
Check		
1½″ and smaller	600 lb ANSI FS screwed, bolted bonnet[4], standard trim	Etc.
2″ and larger	150 lb ANSI CS RF flanged, bolted bonnet,[4] swing check, standard trim	Etc.
Reciprocating Compressor Discharge	300 lb ANSI CS RF flanged, piston check, bolted bonnet[4]	Etc.
Lubricated Plug		
	Not normally used	

(table continued on next page)

Table 9-12
Continued

Size Ranges	General Specifications	Platform Service
Non-Lubricated Plug		
1½″ through 6″	150 lb ANSI CS RF flanged, bolted bonnet	Etc.
Compressor Laterals		
	Use ball valves	
Needle		
¼″ through ½″	6000 lb CWP, bar stock screwed, AISI 316 SS	Etc.
Fittings[5]		
Ells and Tees		
¾″ and smaller	6000 lb FS screwed	ASTM A105-71
1″ through 1½″	3000 lb FS screwed	ASTM A105-71
2″ and larger	Butt weld, seamless, wall to match pipe	ASTM A234, Grade WPB
Unions		
¾″ and smaller	6000 lb FS screwed, ground joint, steel to steel seat	ASTM A105-71
1″ through 1½″	3000 lb FS screwed, ground joint, steel to steel seat	ASTM A105-71
2″ and larger	Use flanges	
Couplings		
1" and smaller	6000 lb FS screwed	ASTM A105-71
1½″	3000 lb FS screwed	ASTM A105-71
Plugs		
1½″ and smaller	Solid bar stock, forged steel	ASTM A105-71
2″ and larger	X-Strong seamless, weld cap	ASTM A234, Grade WPB
Screwed Reducers		
¾″ and smaller	Sch. 160 seamless	ASTM A105-71
1″ through 1½″	Sch. 80 seamless	ASTM A105-71
Flanges[5]		
1½″ and smaller	150 lb ANSI FS RF screwed	ASTM A105-71
2″ and larger	150 lb ANSI FS RF weld neck, bored to pipe schedule	ASTM A105-71
Bolting		
Studs	Class 2 fit, threaded over length	ASTM A193, Grade B7[4]
Nuts	Class 2 fit, heavy hexagon, semi-finish	ASTM A194, Grade 2H[4]
Gaskets	Spiral wound asbestos	Spiral Wound Mfg. Type or___ Mfg. No.___ w/AISI 304 SS windings
Thread Lubricant	Conform to API Bulletin 5A2	Mfg. No.___

NOTES:

1. For glycol service, all valves and fittings shall be flanged or socketweld.
2. See Table 3.1 for pressure-temperature ratings.
3. API 5L, Grade B, Seamless may be substituted if ASTM A106, Grade B, Seamless is not available.
4. Studs and nuts shall be hot-dip galvanized in accordance with ASTM A153.
5. Fittings and flanges that do not require normalizing in accordance with ASTM A105-71, due to size or pressure rating, shall be normalized when used for service temperatures from −20°F to 60°F. Fittings and flanges shall be marked HT, N, * or with some other appropriate marking to designate normalizing.

(Courtesy of API RP 14E)

Table 9-13
Example Pipe, Valves, and Fittings Table

Service: HYDROCARBONS, NON-CORROSIVE, GLYCOL	**PIPING MATERIAL SPECIFICATION SHEET CLASS A** **DESIGN** **Max. Press.** 285 PSIG — **Max. Temp.** 100°F — **Limited By** FLANGES (4)	Engr. Std. Proj. No. Page 1 of 1 Rating: 150#RF, C.S.

SIZE		½	¾	1	1½	2	3	4	6	8	10	12		**Corr. Allow. 0.05"**
PIPE	**SCH 80**							**STD**						
	A-106-B-SMLS.													

Joint Const.	T&C (8)	Butt welded and flanged
Fittings	3000# FS Scrd. A105 (9)	A234 WPB smls. buttweld sch. to match pipe (9)
Flanges	ANSI 150# RF Scrd. A105 (9) (1)	ANSI 150# RFWN (3) A105 (9)
Unions	3000# FS GJ Scrd. Int. seats, A105 (9)	Flanges (3)
Plugs	Hex head scrd., A105 (9)	
Thrd. Nipp & Swages	Sch 160 (5) A-106 (8)	
Ball	VBT-60-1 VBT-60-2 VBS-60-1	VBF-15-1 (6) VBF-15-2 (6)
Check	VCT-60-1 VCS-60-1	VCF-15-1
Gate	VGT-60-1 VGS-60-1	VGF-15-1
Needle	VNT-500-1 VNT-500-2	
Globe	VOT-60-1 VOS-60-1	VOF-15-1
Butterfly		
Diaphragm		
Bolting-STUDS-A193-B7, NUTS-A194-2H CADMIUM PLATED	Gaskets—⅛" FLEXITALLIC CG or Equal	
Miscellaneous—See Instrument Spec. for Size and Type of Instrument Connection		

BRANCH CONNECTIONS

HEADER SIZE (columns) / **BRANCH SIZE** (rows)

BRANCH SIZE			24	20	18	16	14	12	10	8	6	4	3	2	1½	1	¾	½
½			#	#	#	#	#	#	#	#	#	#	#	#	RT	RT	RT	T
¾			#	#	#	#	#	#	#	#	#	#	#	#	RT	RT	T	
1			#	#	#	#	#	#	#	#	#	#	#	#	RT	T		
1½			#	#	#	#	#	#	#	#	#	#	#	#	T			
2			#	#	#	#	#	#	#	#	#	RT	RT	T				
3			W	W	W	W	W	W	W	W	RT	RT	T					
4			W	W	W	W	W	W	W	RT	RT	T						
6			W	W	W	W	W	RT	RT	RT	T							
8			W	W	W	RT	RT	RT	RT	T								
10			W	RT	RT	RT	RT	RT	T									
12			RT	RT	RT	RT	RT	T										
14			RT	RT	RT	RT	T											
16			RT	RT	RT	T												
18			RT	RT	T													
20			RT	T														
24			T															

T — Buttweld, SW; Scr'd Tee
RT — Buttweld, SW; Reducing Tee
— Thredolet; Sock-O-Let (7)
W — Weldolet (Sch to Match Header & Branch) (7)
TR — Straight Tee w/Reducer

NOTES

1.) Use only to match equip. and special items.
2.) Flat face to be used only to match equip.
3.) Flange bore to match pipe I.D.
4.) See ANSI B 16.5 and B 31.3 for pressure and temp. limitations.
5.) All thread pipe nipp. are not acceptable.
6.) Valves listed may require a gear operator. See valve legend for requirement.
7.) TOL, SOL & WOL to be A105
8.) Glycol service piping to be socket welded or buttwelded.
9.) Fittings and flgs shall be normalized and marked when serv. temps. are –20 to 60°F.

(text continued from page 324)

Solution

Criteria

$$V_{max} = 15 \text{ ft/s}$$
$$V_{min} = 3 \text{ ft/s}$$
$$\text{Pressure drop} = 900 - 150 = 750 \text{ psi}$$

Velocity

$$V = \frac{(0.012)(1,030)}{d^2} = \frac{12.36}{d^2}$$

V	ID
3	2.03
15	0.91

Pressure Drop

From Example 8-1, the pressure drop in 2-in. line is acceptable. For flexibility and mechanical strength, it would probably be better to use a 2-in. line than a 1-in. or 1½-in. line.

B 31.3

$$t = \left[0.05 + \frac{(1,480)(2.375)}{2[(20,000)(1) + (1,480)(0.4)]}\right]\left[\frac{100}{100 - 12.5}\right]$$

$t = 0.155$ in.

Can use a standard weight pipe.

B 31.4

$$t = \frac{(1,480)(2.375)}{2(0.72)(1.0)(1.0)(35,000)}$$

$t = 0.0697$ in.

Would probably use a standard weight pipe for mechanical strength.

B 31.8

Outside the facility use $F = 0.72$, wall thickness same as above. Within the facility use $F = 0.6$.

$t = 0.0837$ in.

Example No. 9-2: Gas Line

Given: Same as Example 8-2.
Gas flows to dehydrator, which operates at 800 psi.
Line is rated for 1,480 psi.

Problem: Choose a line size and wall thickness using B 31.3 and B 31.8.

Solution

Criteria

$$V_{max} = 60 \text{ ft/s}$$
$$V_{min} = 10 \text{ to } 15 \text{ ft/s}$$
$$\text{Pressure drop} = 900–800 = 100 \text{ psi}$$

At this low a pressure, erosional velocity is not important.

Velocity

Maximum velocity occurs at the dehydrator.

Z = 0.67 (from Chapter 3)

$$V = \frac{(60)(23)(540)(0.67)}{(815)\,d^2} = \frac{613}{d^2}$$

V	ID
10	7.83
15	6.39
60	3.20

Pressure Drop

From Example 8-2 pressure drop in 4-in. line is not acceptable, but is acceptable in a 6-in. line. This also gives reasonable velocities (between 15 and 60 ft/s).

B 31.3

$$t = \left[0.05 + \frac{(1,480)(6.675)}{2[(20,000)(1) + (1,480)(0.4)]}\right]\left[\frac{100}{100 - 12.5}\right]$$

$t = 0.331$ in.

Use 6-inch XS. Could use 6-inch 0.375 wall if available.

B 31.8

Outside facility:

$$t = \frac{(1,480)(6.675)}{2(0.72)(1.0)(1.0)(35,000)}$$

$t = 0.196$ in.

Use 6-in. std. Could use 6-in. 0.219 wall if available.

Inside facility:

$$t = \frac{(1,480)(6.675)}{2(0.6)(1.0)(1.0)(35,000)}$$

$t = 0.235$ in.

Use 6-in. std.

Example No. 9-3: Two-Phase Line

Given: Same as Example 8-3.
Fluid flows to separator, which operates at 800 psi.
Line is rated for 1,480 psi.

Problem: Choose a line size and wall thickness using B 31.3 and B 31.8.

Solution

Criteria

$$V_{max} = 60 \text{ ft/s or } V_e$$
$$V_{min} = 10 \text{ to } 15 \text{ ft/s}$$
$$\text{Pressure drop} = 100 \text{ psi}$$

Erosional Velocity

$$V_e = \frac{C}{(\rho_m)^{1/2}}$$

Critical condition occurs at lowest pressure but for computing V_e use 900 psi to be conservative.

$\rho_m = 6.93$ lb/ft^3 from Example 8-3.

C	V_e
80	30.4
100	38.0
120	45.6
140	53.2

Minimum ID

Z = 0.67 (from Chapter 3)

$$ID_{min} = \left[\frac{11.9 + \frac{(22,350)\,(540)\,(0.67)}{(16.7)\,(815)}}{1,000\ V}\right]^{1/2} [1,030]^{1/2}$$

$$ID_{min} = \frac{25}{[V]^{1/2}}$$

V	ID_{min}
10	7.9
15	6.5
38	4.1
46	3.7
53	3.4

Pressure Drop

From Example 8-3 a 4-in. line is not acceptable, a 6-in. line is marginal and an 8-in. line is probably okay.

B 31.3

$$t = \left[0.05 + \frac{(1,480)\,(8.625)}{2[(20,000)\,(1) + (1,480)\,(0.4)]}\right]\left[\frac{100}{100 - 12.5}\right]$$

t = 0.411 in.

Use 8-in. XS. Could use 8-in. 0.438 wall if available.

B 31.8

Outside facility:

$$t = \frac{(1,480)\,(8.625)}{2(0.72)\,(1.0)\,(1.0)\,(35,000)}$$

t = 0.253 in.

Use 8-inch, 0.277 wall. With a slight deration, one could use an 8-inch, 0.250 wall.

Inside facility:

$$t = \frac{(1,480)(8.625)}{2(0.6)(1.0)(1.0)(35,000)}$$

$$t = 0.304 \text{ in.}$$

Use 8-in. std. One could use an 8-in., 0.312 wall if available.

REFERENCES

1. Svedeman, Steven J. "Experimental Study of the Erosional/Corrosional Velocity Criterion for Sizing Multiphase Flow Lines," Submitted to American Petroleum Institute, 1993.
2. Wicks, Moye, "Transport of Solids at Low Concentrations in Horizontal Pipes," *Advances in Solid-Liquid Flow in Pipes and its Application,* Iraj Zandi, Pergamon Press, 1971.

CHAPTER

10

Pumps*

INTRODUCTION

Pumps are used in production facilities to move liquid from a low pressure or low elevation location to one of a higher pressure or elevation. Wherever possible it is usually advantageous to locate equipment and select operating pressures in such a manner as to minimize the need for pumping or to minimize the volume for pumping. For example, if produced water is to flow from a CPI to a flotation unit, it is beneficial to locate the flotation unit at a lower elevation and remove the necessity for pumping.

This chapter discusses the classification of pump types, basic principles for selecting pumps, and concepts for pump type selection. Subsequent chapters discuss in more detail the various types of centrifugal and reciprocating pumps and the details of their construction.

PUMP CLASSIFICATION

Pumps are classified as either "kinetic" or "positive displacement" pumps. In a kinetic pump, energy is added continuously to increase the fluid's velocity within the pump to values in excess of those that exist in

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the discharge pipe. Passageways in the pump then reduce the velocity until it matches that in the discharge pipe. From Bernoulli's law, as the velocity head of the fluid is reduced, the pressure head must increase. Therefore, in a kinetic pump the kinetic or velocity energy of the fluid is first increased and then converted to potential or pressure energy. Almost all kinetic pumps used in production facilities are centrifugal pumps in which the kinetic energy is imparted to the fluid by a rotating impeller generating centrifugal force.

In a positive displacement pump the volume containing the liquid is decreased until the resulting liquid pressure is equal to the pressure in the discharge system. That is, the liquid is compressed mechanically, causing a direct rise in potential energy. Most positive displacement pumps are reciprocating pumps where the displacement is accomplished by linear motion of a piston in a cylinder. Rotary pumps are another common type of positive displacement pump, where the displacement is caused by circular motion.

CENTRIFUGAL PUMPS

Centrifugal pumps are classified as either radial flow or axial flow. Figure 10-1 shows a radial flow pump. Flow enters the center of the rotating wheel (impeller) and is propelled radially to the outside by cen-

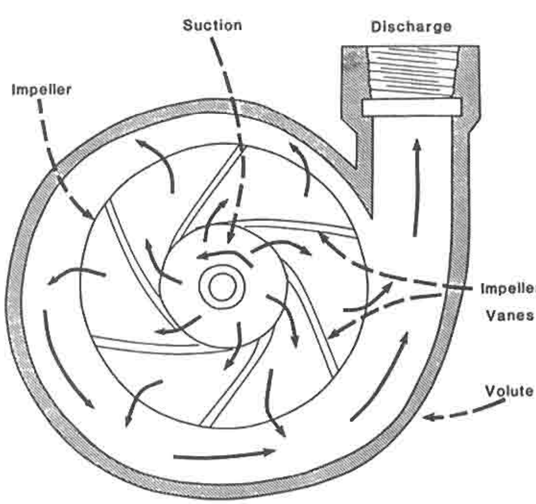


Figure 10-1. Radial flow pump.

trifugal force. Within the impeller the velocity of the liquid is increased, and this is converted to pressure by the case.

A typical axial flow pump is shown in Figure 10-2. Flow is parallel to the axis of the shaft. A velocity is imparted by the impeller vanes, which are shaped like airfoils.

Most pumps are neither radial flow nor completely axial flow but have a flow path somewhere in between the two extremes. Radial flow pumps develop a higher head per stage and operate at slower speeds than axial flow pumps. Therefore, axial flow designs are used in very high flow rate, very low head applications.

Figure 10-3 shows a typical head-capacity curve for a centrifugal pump. At a constant speed (i.e., rotational velocity), as the head required to be furnished by the pump efficiency curve. For a given impeller shape, the efficiency is a maximum at a design throughput rate. As the rate varies upward and downward from this point the efficiency decreases.

By varying the pump speed the throughput at a given head or the head for a given throughput can be changed. In Figure 10-4 as the speed decreases from N_1 to N_2 to N_3, the flow rate decreases if the head required is constant, or the head decreases if the flow rate is constant.

In most piping systems both the head and the flow rate vary because the system has its own required pump head for a given flow rate. This can be seen by the example in Figure 10-5 where the head required by the system for the pump to provided is merely the friction drop in the pipeline between points A and B. This is a function of flow rate and can

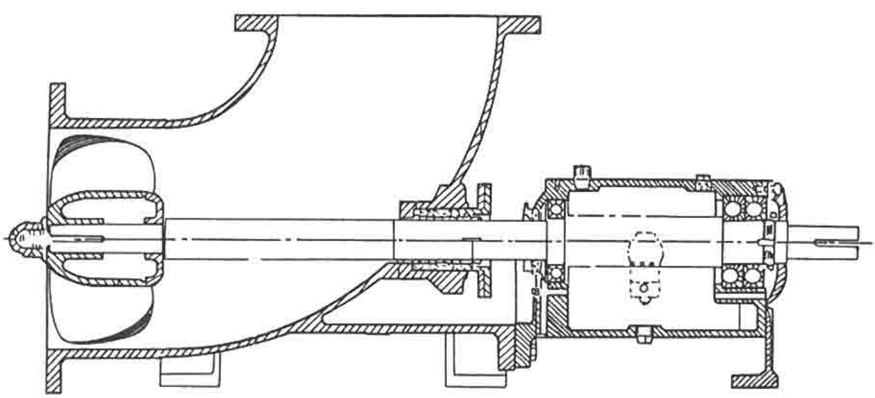


Figure 10-2. Axial flow pump.

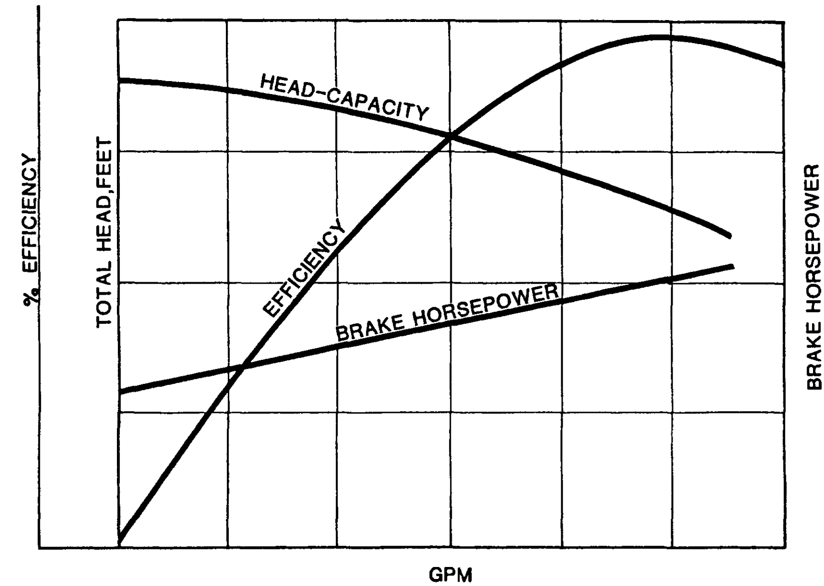


Figure 10-3. Centrifugal pump performance curves.

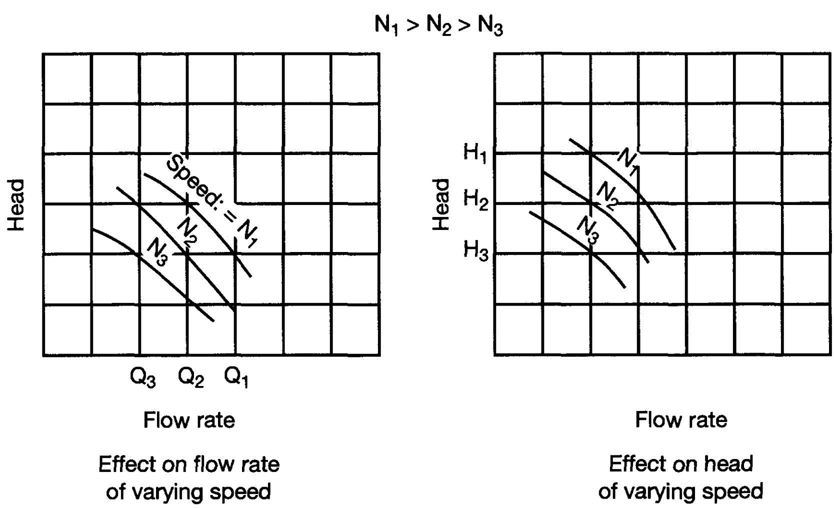


Figure 10-4. Effect of pump speed on a centrifugal pump.

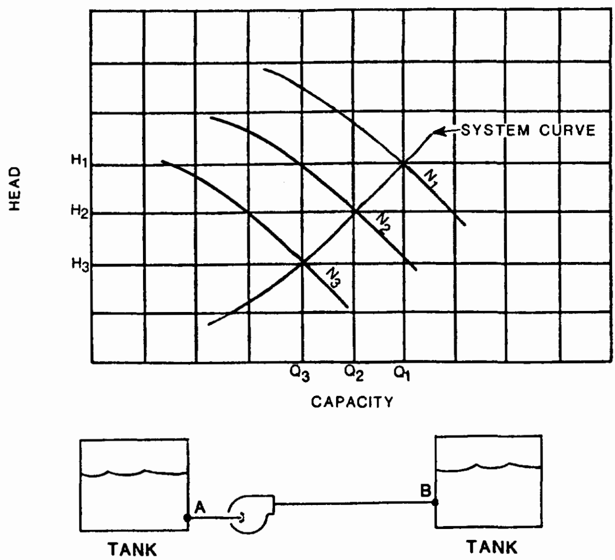


Figure 10-5. System-pump interaction.

therefore be plotted as a "system curve" on the pump-head-flow-rate curve. For this system, as the pump is speeded up or slowed down a new equilibrium of head and flow rate is established by the intersection of the system curve with the pump curve.

Figure 10-6 shows how the throughput can be changed by imposing an artificial backpressure on the pump. By adjusting its orifice, the control valve can shift the system curve, establishing new head-flow-rate equilibria. As the pressure drop across the control valve increases from ΔP_1 to ΔP_2 to ΔP_3 the flow rate through the system decreases from Q_1 to Q_2 to Q_3.

The advantages of centrifugal pumps are:

1. They are relatively inexpensive.
2. They have few moving parts and therefore tend to have greater onstream availability and lower maintenance costs than positive displacement pumps.
3. They have relatively small space and weight requirements in relation to throughput.
4. There are no close clearances in the fluid stream and therefore they can handle liquids containing dirt, abrasives, large solids, etc.

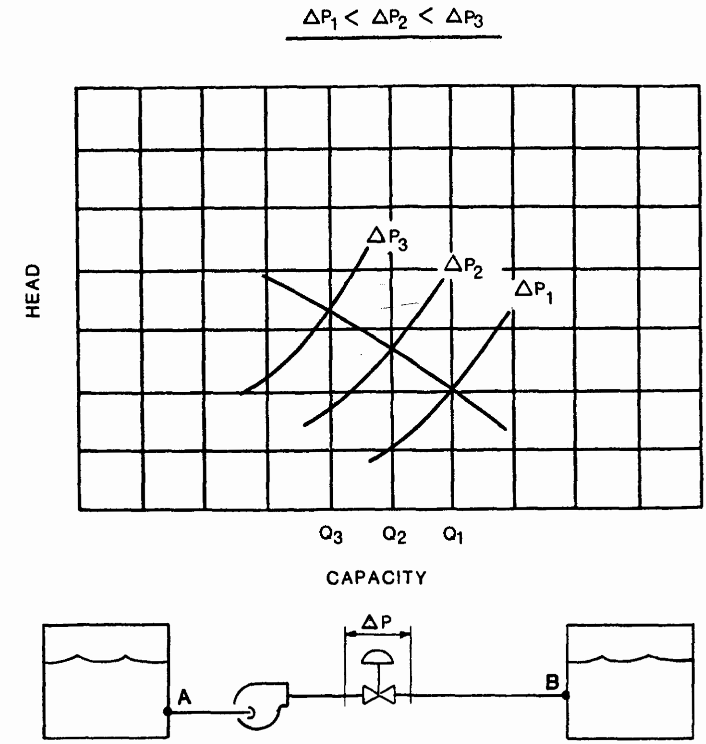


Figure 10-6. System-pump interaction.

5. Because there is very little pressure drop and no small clearances between the suction flange and the impeller, they can operate at low suction pressures (we will define a term, "net positive suction head," shortly).
6. Due to the shape of the head-capacity curve, centrifugal pumps automatically adjust to changes in head. Thus, capacity can be controlled over a wide range at constant speed.

Although several impellers can be installed in series to create large heads, centrifugal pumps are only practical for achieving high pressure when there are large flow rates. In addition centrifugal pumps have low maximum efficiencies when compared to reciprocating pumps. Since the efficiency also declines as the flow rate varies from the design point, in actual operation, the pump will operate at still lower efficiencies. Efficiencies between 55% and 75% are common.

RECIPROCATING PUMPS

In reciprocating pumps, energy is added to the fluid intermittently by moving one or more boundaries linearly with a piston, plunger, or diaphragm in one or more fluid-containing volumes. If liquid is pumped during linear movement in one direction only then the pump is classified "single acting." If the liquid is pumped during movement in both directions it is classified as "double acting."

Figure 10-7 shows both a single-acting and a double-acting pump. As the plunger, A, moves to the right in the single acting pump the fluid is compressed until its pressure exceeds the discharge pressure and the discharge check valve, B, opens. The continued movement of the plunger to the right pushes liquid into the discharge pipe. As the plunger begins to move to the left, the pressure in the cylinder becomes less than that in the discharge pipe and the discharge valve closes. Further movement to the

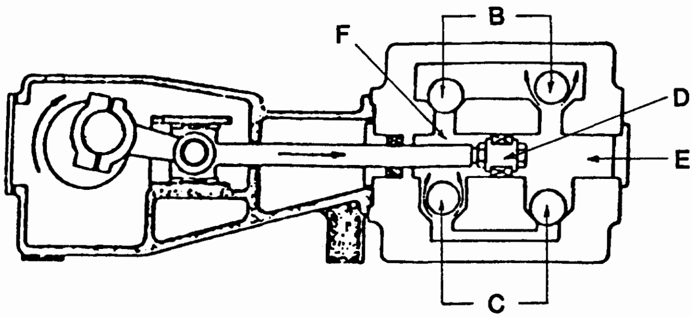


Figure 10-7a. Double-acting piston pump.

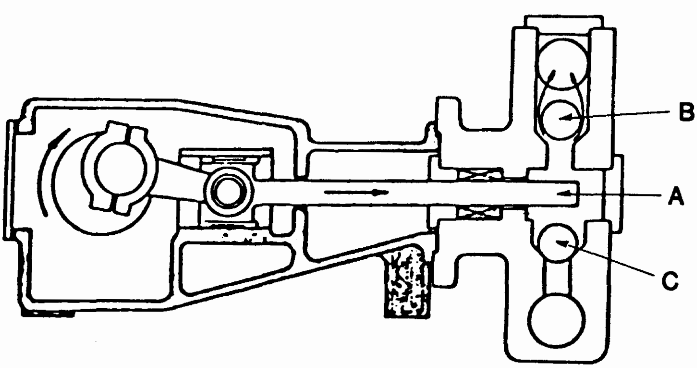


Figure 10-7b. Single-acting plunger pump.

left causes the pressure in the cylinder to continue to decline until it is below suction pressure. At this point the suction check valve, C, opens. As the plunger continues to move to the left the cylinder fills with liquid from the suction. As soon as the plunger begins to move to the right it compresses the liquid to a high enough pressure to close the suction valve and the cycle is repeated. Thus, liquid is discharged only when the plunger moves to the right.

In a double-acting pump the plunger is replaced by a piston, D. When the piston moves to the right, the liquid in the cylinder to the right of the piston, E, is discharged and the cylinder to the left of the piston, F, is filled. When the direction of the piston is reversed the liquid in F is discharged and the cylinder at E is filled with suction fluid. Thus, liquid is pumped when the cylinder moves in either direction.

Reciprocating pumps are also classified by the number of cylinders they have. If the liquid is contained in one cylinder it is called a simplex pump, two cylinders a duplex, three cylinders a triplex, five cylinders a quintuplex, seven cylinders a septuplex, and so forth.

The head-flow-rate curve is a nearly straight vertical line. That is, no matter how high a head is required, the plunger will displace a given volume of liquid for each rotation. The flow rate through the pump can only be varied by changing the pump speed. A throttling valve that changes the system head-flow-rate curve will have no effect on the flow rate through the pump.

Since the pump will attempt to match whatever system pressure is required, it is necessary that a relief valve be installed on the pump discharge. This assures that the pump will not overpressure itself or the discharge pipe.

The advantages of reciprocating pumps are:

1. The efficiency is high regardless of changes in required head. Efficiencies on the order of 85% to 95% are common.
2. The efficiency remains high regardless of pump speed, although it tends to decrease slightly with increasing speed.
3. Reciprocating pumps run at much lower operating speeds than centrifugal pumps and thus are better suited for handling viscous fluids.
4. For a given speed the flow rate is constant regardless of head. The pump is limited only by the power of the prime mover and the strength of the pump parts.

Because of the nature of their construction, reciprocating pumps have the following disadvantages when compared to centrifugal pumps:

1. They have higher maintenance cost and lower availability because of the pulsating flow and large number of moving parts.
2. They are poorer at handling liquids containing solids that tend to erode valves and seats.
3. Because of the pulsating flow and pressure drop through the valves they require larger suction pressures (net positive suction head) at the suction flange to avoid cavitation.
4. They are heavier in weight and require more space.
5. Pulsating flow requires special attention to suction and discharge piping design to avoid both acoustical and mechanical vibrations.

In reciprocating pumps, the oscillating motion of the plungers creates disturbances (pulsations) that travel at the speed of sound from the pump cylinder to the piping system. These pulsations cause the pressure level of the system to fluctuate with respect to time. In order to lessen potentially damaging pulsations in piping from this pressure fluctuation, pulsation dampeners may be installed in the suction and/or discharge piping of the reciprocating pump. Pulsation dampeners are recommended for all major multi-pump installations, unless computer analog studies indicate that they are not needed. There are basically four types of pulsation dampeners: liquid-filled, gas-liquid interface, gas-cushioned, and tuned acoustical filter.

Liquid-filled dampeners (Figure 10-8a) are large surge vessels located close to the pump. The volume of the vessel is normally ten times the pump throughput per minute. Liquid-filled dampeners are maintenance-free and incur no pressure drop. They are heavy when filled with liquid, however, and they tend to require a lot of space.

A gas-liquid interface pulsation dampener (Figure 10-8b) consists of a vessel that is partially filled with liquid and partially filled with gas. This pulsation dampener is effective because the high compressibility of the gas absorbs the pressure pulses. Due to the compressibility of the gas, these dampeners are much smaller than liquid-filled dampeners. However, since the gas could be dissolved slowly in the liquid that is being pumped, the gas-liquid interface must be monitored. In addition, these dampeners are impractical at locations where pressures vary widely.

Gas-cushioned pulsation dampeners (Figure 10-8c) use a pressure bladder to keep gas from being absorbed in the liquid. These dampeners

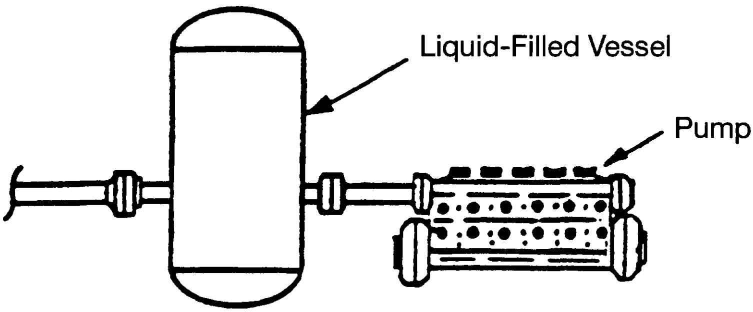


Figure 10-8a. Liquid-filled pulsation dampener.

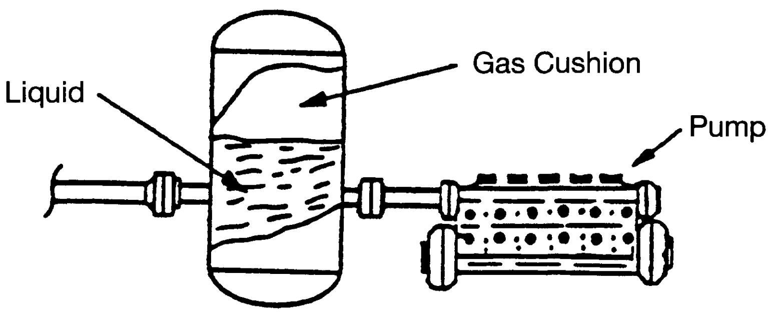


Figure 10-8b. Gas-liquid interface pulsation dampener.

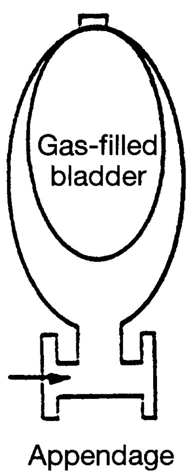


Figure 10-8c. Gas-cushioned pulsation dampener.

are generally small and inexpensive, but they cannot be used in applications where temperatures are above 300°F. In addition, like gas-liquid interface dampeners, these dampeners are impractical at locations where pressures vary greatly.

Tuned acoustical filters (Figures 10-8d and 10-8e) are generally of two types: Type I and Type II. In Type I, two liquid-filled dampeners are connected by a short section of small-diameter pipe called a choke tube; in Type II, the choke tube is installed between the one liquid-filled dampener and the piping system. Both dampening systems are best designed through the use of digital or analog simulation. Tuned acoustical filters can suppress pulsations at all frequencies. In addition, they incur low maintenance costs, and their performance is not affected by temperature. Still, these filters require ample space and are relatively expensive. They also require a pressure drop across the choke tube, thereby limiting applicability in low-NPSH situations.

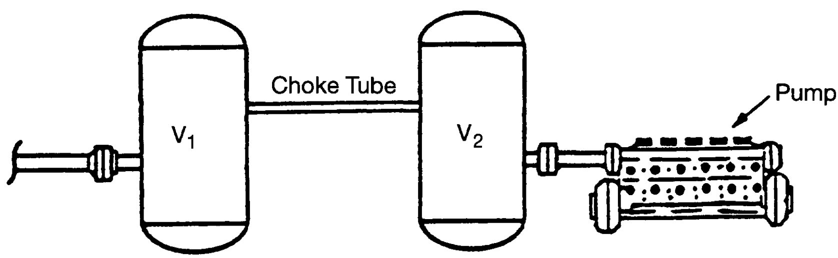


Figure 10-8d. Type I tuned acoustical filter.

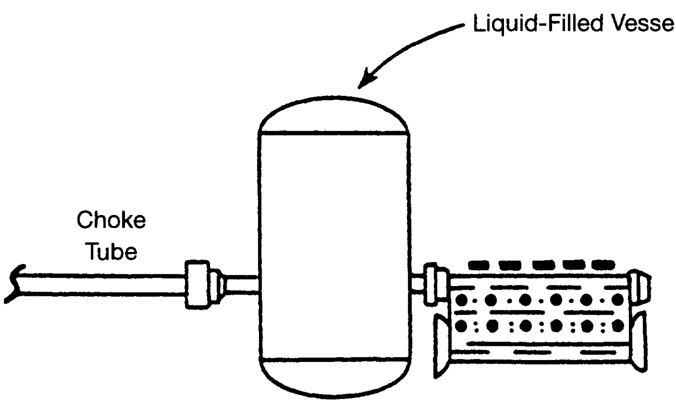


Figure 10-8e. Type II tuned acoustical filter.

DIAPHRAGM PUMPS

These pumps are a special type of reciprocating pump that utilize the action of a diaphragm moving back and forth within a fixed chamber. Sometimes the diaphragm is used to power a reciprocating pump with air or natural gas. Figure 10-9 shows a typical diaphragm pump where flexure of the diaphragm creates the pumping action. When gas pressure is applied against either diaphragm it forces liquid out. When the gas is relieved the diaphragm flexes under the pressure in the suction line and allows liquid to enter.

The advantages of a diaphragm pump are that it can handle large amounts of suspended solids, is inexpensive to repair, can handle low flow rates inexpensively, and can run periodically without any liquid. However, diaphragm pumps require frequent maintenance because they are reciprocating pumps and because the diaphragm has a tendency to fatigue with time. They generally cannot handle very high flow rates, or discharge pressures. A patent-pending high-pressure diaphragm pump is currently being offered by a leading diaphragm pump manufacturer.

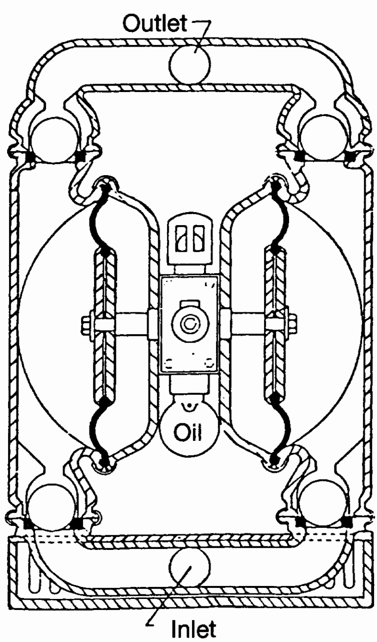


Figure 10-9. Diaphragm pump.

Claims of discharge pressures of up to 250 psig (with 80 psig air/gas supply) have been advertised, and flow rates of up to 96 gpm are reported. High-pressure output is obtained through the use of a surface area differential to intensify pressure output. When a diaphragm action is used to power a reciprocating plunger pump it is possible to handle large discharge pressures, but only if the flow rate is very small. Be careful that any mixing of power gas and pumped fluid does not create a hazard in case there is leakage through the diaphragm.

ROTARY PUMPS

These pumps operate by having a rotating member turn inside a housing in such a way as to create trapped liquid through the pump. Figure 10-10 shows several configurations of rotary pumps. Although these pumps may look like centrifugal pumps, their action is that of a positive displacement pump in that the liquid is continually compressed to a high pressure without first being given a high kinetic energy.

Rotary pumps have the same characteristics as reciprocating pumps, except that at low speed leakage between the cavities increases. At very low speeds the reduction in efficiency can be very significant. When compared to reciprocating pumps, rotary pumps require less space, and deliver relatively pulsation-free flow. Their main advantage is that unlike reciprocating and centrifugal pumps, their construction subjects the pumped fluid to a minimum amount of shear or turbulence. Thus, they tend to be used in process applications where one of the other pump types could be expected to shear and disperse one liquid into another making subsequent treating more difficult.

Their disadvantages are that they have close clearances that require that the liquids being pumped have a lubricating value, be non-corrosive, and contain few solids. Therefore, they tend to be limited to relatively solids-free oil or emulsion streams.

In addition to the standard rotary pumps shown in Figure 10-10, another type of positive displacement rotary pump exists in which the pump, motor, and pumped liquid are completely contained within a closed vessel (Figure 10-11). This type of pump is known as a "canned" rotary pump since the motor/pump package is contained within the closed vessel, or "can." This type of pump is essential for pumping toxic liquids, radioactive waste water, and other liquids that pose serious risks if a shaft seal failure occurs. Although canned centrifugal pumps have existed for many years,

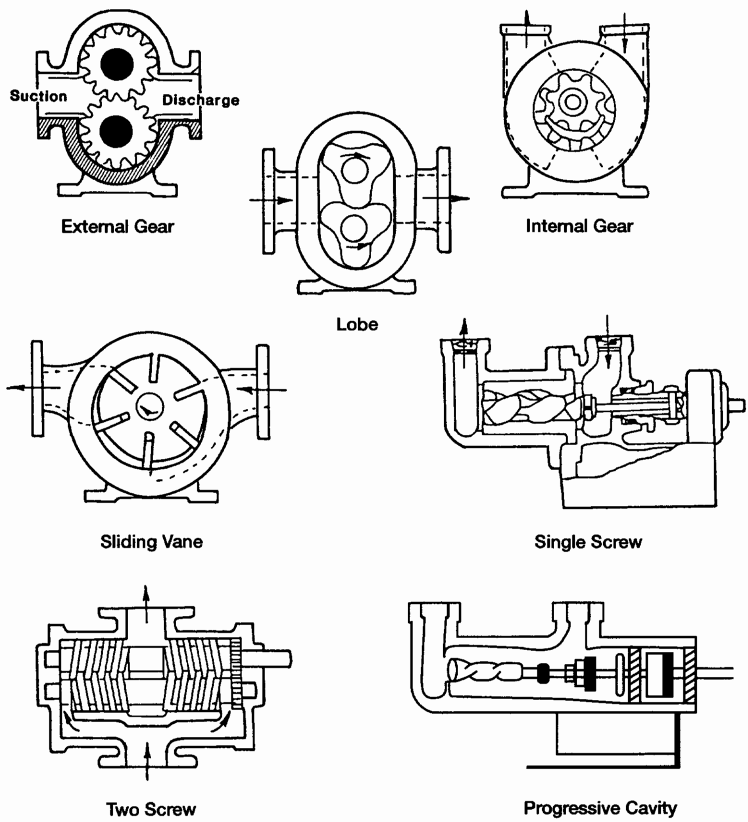


Figure 10-10. Typical types of rotary pumps.

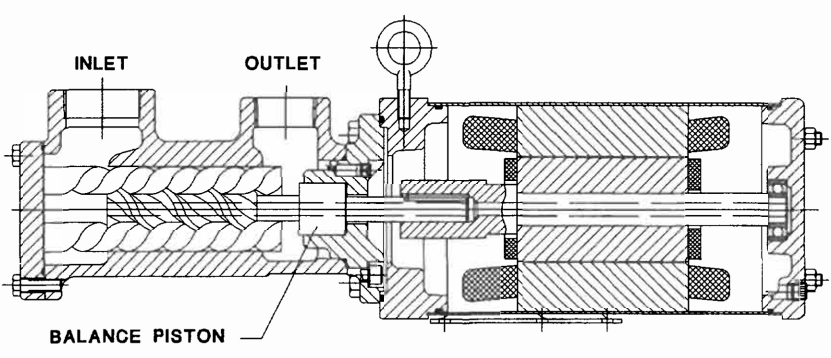


Figure 10-11. Canned rotary pump (*courtesy of Imo Pump*).

they were inefficient and—in some cases—not able to reasonably handle more viscous fluids. Today's canned rotary pumps are able to accommodate highly viscous fluids. As for pricing, the current canned motor/pump package compares well with conventional shaft-sealed pumps that are flexibly coupled to standard industrial three-phase AC motors.

MULTIPHASE PUMPS

Multiphase fluids typically produced from an oil well consist of hydrocarbon liquid, hydrocarbon gas, and an immiscible water phase. These fluids historically must be processed by a multiphase production system near the wells. This arrangement is needed because transfer of the multiphase fluids is achieved through the use of reservoir energy and, in most cases, this energy is insufficient to transfer fluids over any considerable distance. The inherent problem with processing multiphase fluids close to the wells is the high capital and operating cost experienced (both onshore and offshore). A solution to this problem is to obtain a pump that can handle unprocessed multiphase fluids and transport them over considerable distances; the multiphase pump can do this task. This pump is able to boost the pressure of wellhead fluids over considerable distances to a central processing facility, thereby eliminating several smaller local processing facilities. In addition to providing for economic savings from consolidation of surface and offshore facilities, the use of multiphase pumps makes the development of satellite fields more economically attractive. Multiphase pumps also aid in increasing well production rates by lowering required back-pressure on wells.

Multiphase pumps are applicable in services where the gas volume fraction (GVF) is as high as 95%. To determine the GVF, divide the actual gas flow rate by the total mixed flow rate. For a GVF above 95%, the volumetric efficiency decreases, and more fluids (gas) slip back to the pump inlet. Slower speed and higher pressure boost also increase slip, and thus decrease volumetric efficiency. A typical twin-screw multiphase pump is shown in Figure 10-12.

BASIC PRINCIPLES

Head

The pressure that a pump must put out is usually expressed in head, or the pressure generated by an equivalent height of liquid. The head

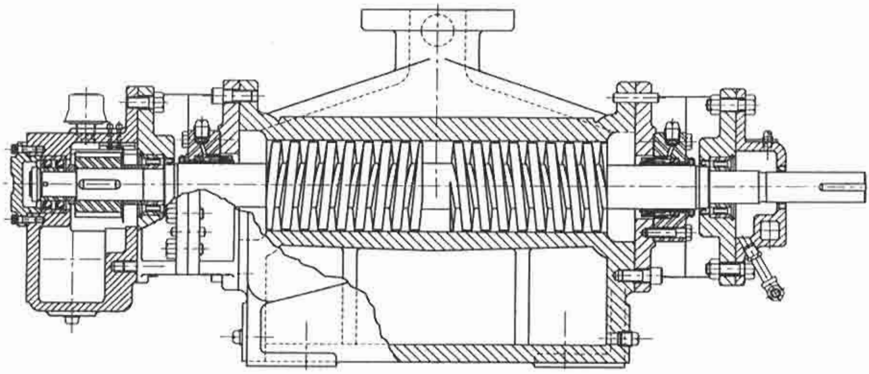


Figure 10-12. A twin-screw multiphase pump (*courtesy of Ingersoll-Dresser Pumps*).

required to pump a fluid between two points in a piping system can be calculated by rearranging Bernoulli's law:

$$H_p = H_2 + H_f - H_1 \tag{10–1}$$

where H_p = head required for the pump, ft
H_1 = total fluid head (elevation plus pressure plus velocity) at point 1, ft
H_2 = total fluid head at point 2, ft
H_f = head lost due to friction between points 1 and 2, ft

By substituting for each of the terms in total fluid head and rearranging terms:

$$H_p = \frac{144}{\rho}(P_2 - P_1) + (Z_2 - Z_1) + \frac{(V_2^2 - V_1^2)}{2g} + H_f \tag{10-2}$$

where ρ = density of the fluid, lb/ft^3
P_1, P_2 = pressure, psi
Z_1, Z_2 = elevation, ft
V_1, V_2 = velocity, ft/sec
g = 32.2 ft/sec^2

In most pumping installations the difference in velocity head can be ignored. If Equation 10-2 indicates $H_p = 0$, then the piping is in equilibrium and no pump energy is needed to obtain the desired flow. If $H_p < 0$,

this means that the system is not in equilibrium and flow will increase, causing H_f to increase until $H_p = 0$ and equilibrium is established. If increased flow is not desirable, a restriction must be placed in the pipe to cause an artificially high friction drop.

Horsepower

The hydraulic horsepower that must be developed by the pump is given by:

$$\text{HHP} = \frac{H_p \rho Q}{550} \tag{10-3}$$

where HHP = hydraulic horsepower where one horsepower equals 550 ft-lb/sec
H_p = pump head, ft
ρ = density of liquid, lb/ft^3
Q = flow rate, ft^3/sec

By making the appropriate unit conversions Equation 10-3 may be expressed as:

$$\text{HHP} = \frac{(\text{S.G.})\, q'\, H_p}{3{,}960} \tag{10-4}$$

$$\text{HHP} = \frac{q'\, \Delta P}{1{,}714} \tag{10-5}$$

$$\text{HHP} = \frac{Q_1\, \Delta P}{58{,}766} \tag{10-6}$$

where S.G. = specific gravity relative to water
q' = flow rate, gpm
Q_1 = flow rate, bpd
ΔP = pressure increase, psi

The input horsepower to the shaft of the pump is called the brake horsepower, and it is given by:

$$\text{BHP} = \frac{\text{HHP}}{E} \tag{10-7}$$

where BHP = brake horsepower
E = pump efficiency

Net Positive Suction Head (NPSH)

Each pump requires a certain minimum pressure at its suction flange to assure that no vapor is flashed between the pump suction and the cylinder or entrance to the impeller vane. If a lower pressure is supplied, vapor could be liberated by the liquid in the form of small bubbles that would then collapse in an "implosion" as the liquid is pressured in the pump. This is called cavitation and results in noise, vibration, greatly increased wear, and reduced pressure or throughput capacity.

The definition of NPSH is the *net* pressure above the vapor pressure of the liquid being pumped. When a liquid's pressure falls below its vapor pressure, gas if flashed. Since cavitation is first and foremost the formation of gas, we can avoid cavitation by assuring that the liquid's pressure does not drop below its vapor pressure anywhere as it passes through the pump.

The minimum pressure required at the pump flange is expressed in feet of liquid and is called the Net Positive Suction Head (NPSH). It is specified by the manufacturer for each pump. It is determined by measuring the NPSH at the flange, which results in a 3% reduction in throughput capacity when the pump is handling water at 60°F. When the water is flowing at a higher temperature or when hydrocarbons are flowing, it is possible to recalculate the reduced NPSH requirement. Reductions to as low as 10 ft or 50% of that required for cold water are possible. It should be remembered that while these reductions are theoretically permissible, they should not be counted on for most practical design applications because they are not precise and because they do provide a type of safety margin to the design.

The required NPSH is higher for reciprocating pumps due to valve and fluid acceleration losses within the pump. Because the effects of cavitation can be severe, it is recommended that pumps be specified with a required NPSH that is 3 to 5 ft less than the NPSH available from the system.

The NPSH available from the system can be calculated from:

$$NPSH_a = H_p - H_{vpa} + H_{st} - H_f - H_a \quad (10\text{–}8)$$

where $NPSH_a$ = net positive suction head available, ft
H_p = absolute pressure head on surface of liquid in feed tank, ft
H_{vpa} = absolute vapor pressure of liquid at flow conditions, ft
H_{st} = static head of inlet above pump centerline, ft

H_f = friction head loss, ft
H_{vh} = velocity head, ft

$$= \frac{V^2}{2g}$$

H_a = acceleration head, ft

The relationship between H_p and H_{vpa} depends upon the process conditions to which the liquid has been subjected. In most oilfield processes the liquid has always been in equilibrium with a gas phase and no matter what the pressure is on the surface of the feed tank any lowering of pressure would cause gas to flash. Thus, in these cases $H_p = H_{vpa}$. In some instances the liquid has already been flashed at a lower pressure and the process has kept gas from being redissolved in the liquid when the pressure is increased. In these cases $H_p > H_{vpa}$.

The velocity head is normally small in relation to the other factors and can be ignored. The friction head loss can be calculated from pressure drop considerations. The acceleration head is zero for centrifugal pumps. For reciprocating pumps it can be calculated from:

$$H_a = \frac{L\,V\,R_p C}{g\,K} \tag{10-9}$$

where L = actual length of suction pipe, ft
V = average liquid velocity, ft/sec
R_p = pump speed, rpm
C = pump type factor
= 0.200 simplex double-acting
= 0.200 duplex double-acting
= 0.115 duplex single-acting
= 0.066 triplex
= 0.040 quintuplex
= 0.028 septuplex
K = compressibility factor
= 1.4 no compressibility
= 1.5 amine, lean glycol, produced water
= 2.0 crude oil
= 2.5 relatively compressible liquids (e.g., hot oil, ethane)

BASIC SELECTION CRITERIA

In order to make a selection of the pump required for a specific installation it is necessary to first determine the desired flow rate or head. The NPSH available should be determined, and if a centrifugal selection is possible, a system head-flow-rate curve should be developed.

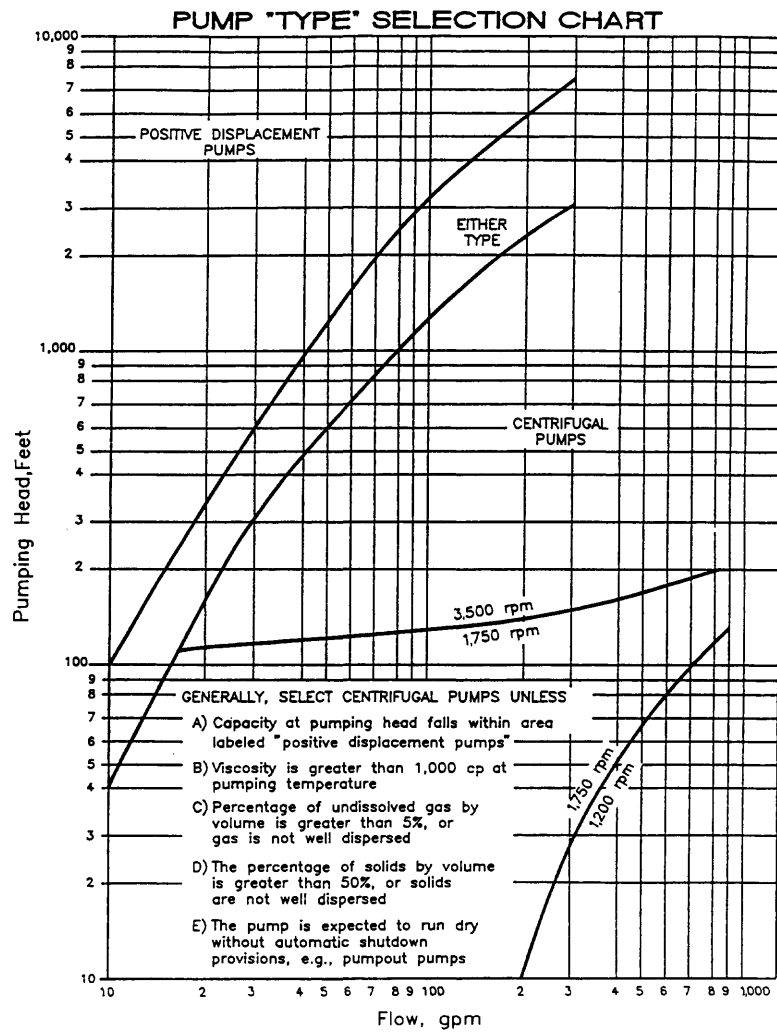
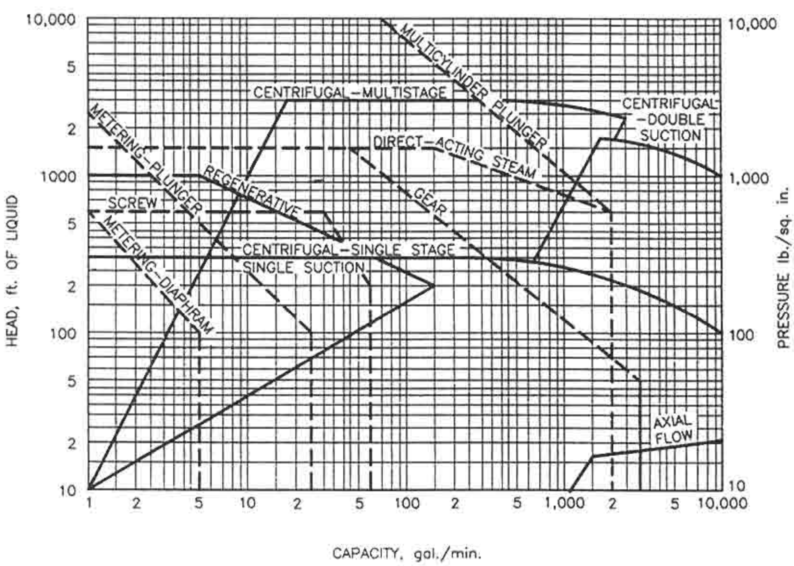


Figure 10-13. Guide for selecting the most economical pump.

Generally, positive displacement pumps are better suited for high head and low flow-rate applications. Figures 10-13 and 10-14 can be used as guides for the type of pump that will probably be found to be most economical. Naturally, the economics and thus the choice of pump type will vary from installation to installation. Figures 10-13 and 10-14 should be used merely to provide general guidance and not to justify any one specific decision.

In choosing a pump type it is necessary to consider the physical constraints of the system. Space availability or weight limitations may dictate a centrifugal or rotary pump; the need to pump solids may dictate a centrifugal or rotary pump; the need to pump solids may dictate a centrifugal or diaphragm pump; and the need to run dry may dictate a diaphragm pump. Where discharge pressures vary over a large range, but flow must remain constant, a positive displacement pump is probably a good choice.

In many high-head applications requiring a reciprocating pump the NPSH available may not be sufficient for a straightforward pump choice.



PUMP COVERAGE CHART BASED ON NORMAL RANGES OF OPERATION OF COMMERCIALLY AVAILABLE TYPES. SOLID LINES—USE LEFT ORDINATE, HEAD SCALE. BROKEN LINES—USE RIGHT ORDINATE, PRESSURE SCALE.

Figure 10-14. Pump selection guide (*from Gas Processors Suppliers Association*).

In these cases low NPSH centrifugal pumps are sometimes used as "charge" pumps to feed the suction of the reciprocating pump.

The number of pumps required for a given installation depends on a balance of capital cost and operating flexibility. For most small installations a choice must be made whether the cost of a standby pump is justified by the potential foregone income if the pump must be shut down for maintenance. On larger installations several other alternatives should be investigated. These could include:

1. One pump rated at 100% throughput.
2. Two pumps rated at 100% throughput each (one pump is standby).
3. Two pumps rated at 50% throughput each (if one pump is down, throughput is decreased to 50% of design).
4. Three pumps rated at 50% throughput each (two operate, one is standby).
5. Three pumps rated at 33% throughput each.

In making the selection it is important to consider the actual throughput requirements over the life of the installation and not merely the peak design throughputs. Finally, the choice should consider the availability of spare parts and service at the location and the preferences of operating personnel.

CHAPTER

11

Centrifugal Pumps*

INTRODUCTION

This chapter discusses practical aspects of centrifugal pump selection. It covers the main operating characteristics that affect these pumps' performance. Also covered are topics such as multiple pump installations, specific speed, the generic types of centrifugal pumps in common oilfield use, the differences between the ANSI and API standards, bearing, seal and wear ring options, and installation considerations. The purpose of this chapter is to familiarize the project engineer with the major options and choices available. There is much more detail that could be written about centrifugal pump design and selection of manufacturing detail. On large complex installations this is best left to a mechanical engineer who is knowledgeable in this area and aware of the current specific detail differences between manufacturers. However, the information in this chapter should provide the information that the project engineer must know if he is going to analyze vendor proposals on simple jobs, or communicate effectively with mechanical engineers on more complex installations.

*Reviewed for the 1998 edition by Fernando C. De La Fuente of Paragon Engineering Services, Inc.

MULTIPLE PUMP INSTALLATIONS

In designing multiple centrifugal pump installations it is necessary to keep in mind the interaction between the pump curves and the system curve. That is, throughput cannot be doubled by adding an identical pump in parallel, and head is not doubled by adding an identical pump in series.

The effect of adding two identical pumps in parallel can be seen in Figure 11-1. Curve A is the pump curve for one pump. Curve B is constructed by doubling the flow rate at a given head to show how the pumps behave in parallel operation. Curve C shows a system curve where the addition of the second pump adds only about 50% to system throughput. Curve D shows a steeper system curve where the system throughput is only increased about 20%.

Figure 11-2 shows the effect of installing two pumps in series. Curve A is the head-flow-rate curve for one pump. The combined curve for both pumps, B, is constructed by doubling the head of Curve A at each value of flow rate. The benefit of the additional pump can be seen by inspecting the intersection of the system curves, C and D, with the pump curves.

The choice of whether to add an additional pump in series or in parallel is illustrated by Figure 11-3. If the system curve is shallow, more throughput is obtained from parallel operation. If the system curve is steep, more throughput can be obtained by series installation.

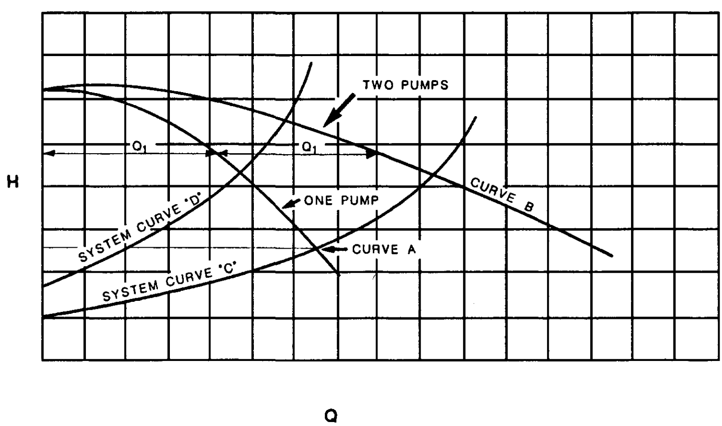


Figure 11-1. Two identical pumps in parallel.

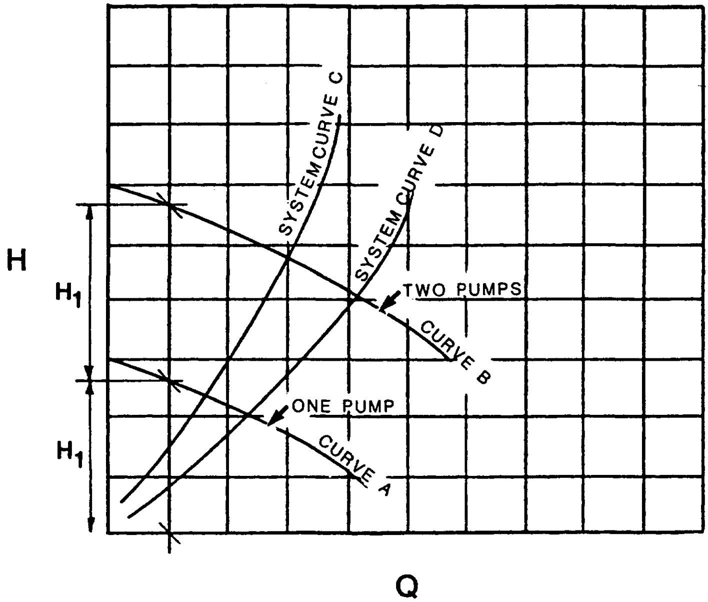


Figure 11-2. Two identical pumps in series.

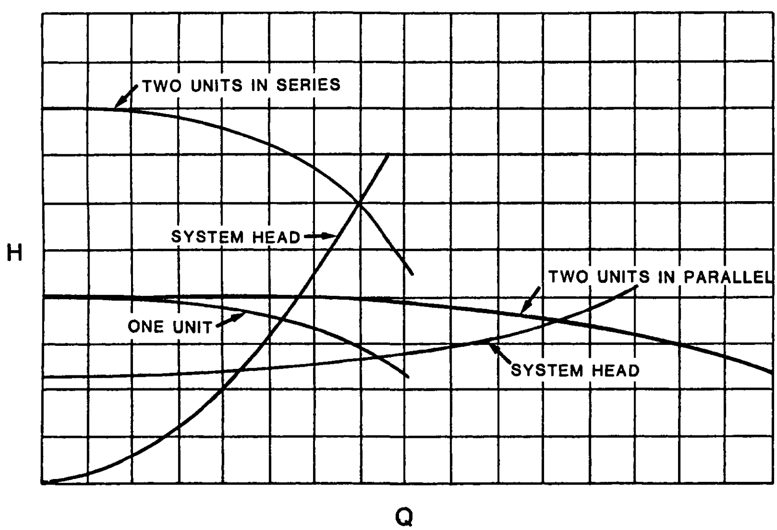


Figure 11-3. Series vs. parallel operation.

PUMP SPECIFIC SPEED

In comparing similar centrifugal pumps it is often useful to use a parameter called specific speed. For similar designs major pump dimensions are proportional to specific speed. Thus, if the performance curve is known for one pump, it can be estimated for another pump of similar design but different impeller diameter.

The specific speed is given by the following formula:

$$N_s = \frac{R_p\,(q')^{1/2}}{H_p^{\,3/4}} \tag{11-1}$$

where N_s = pump specific speed
R_p = pump speed, rpm
q' = flow rate, gpm
H_p = pump head, ft

A pump's specific speed is always calculated at its point of maximum efficiency. It is not a dimensionless number, so that it is critical that the units used in calculating the specific speed be known.

Specific speed is used by pump designers to help determine the required impeller geometry. The lower the specific speed, the more the impeller shape will approach true radial flow. The higher the specific speed, the more closely the impeller needs to approach true axial flow.

Specific speed is also useful in estimating maximum attainable pump efficiency. This is done from Figure 11-4, which is published by the Hydraulic Institute. This figure is useful to help estimate brake horsepower and to help validate vendor's quoted efficiencies.

CODES AND STANDARDS

The two common codes used for centrifugal pumps are API-610 "Centrifugal Pumps for General Refinery Services" and ANSI B73.1 "Specifications for Horizontal, End Suction Centrifugal Pumps for Chemical Process." The API is more stringent in design requirements and quality control and is normally used for critical services where reliability is important.

Table 11-1 shows a comparison of some of the major requirements of these two standards. It can be seen that the API requirements are more stringent. ANSI pumps, or "ANSI type" pumps for pressure ratings in excess of the 150 Class, are less expensive and much more readily avail-

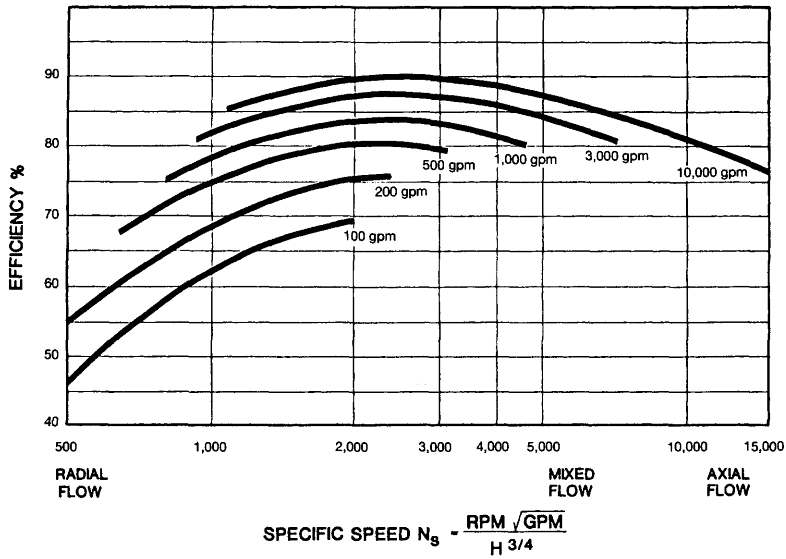


Figure 11-4. Maximum pump efficiency.

able. Wherever service conditions allow, considerable savings and time are possible by specifying an ANSI pump.

GENERIC TYPES OF CENTRIFUGAL PUMPS

Centrifugal pumps are separated into generic types according to their physical construction, number of impellers, or the standard under which they are constructed. The common types and their applications are described below:

ANSI Pump (Figure 11-5)

Description

Mounting = Horizontal

Casing split = Radial

Impeller type = Radial

Mounting feet = Bottom of pump casing

No. of stages = Single

Table 11-1
Comparison of API and ANSI Pump Standards

	API-610	ANSI B73.1
Pressure Rating	• All pressures and temperatures normally encountered	• ANSI Class 150 and temperatures up to 500°F
Pump Casings	• ASME Section VIII, Div. 1 • Carbon or alloy steel required in flammable or toxic service	• Ductile iron, carbon steel or alloy steel required for flammable or toxic service
Impellers	• Single piece castings • Secured to shaft with a key	• May be keyed or threaded to shaft
Wear Rings	• Minimum hardness of 400 BHN or hardness difference of 50	• No requirements
Mechanical Seals	• Required	• No requirements
Shaft Critical Speed	• Lateral critical speed greater than 120% of maximum pump speed	• No requirements
Bearings	• Three-year life for ball bearings • Hydrodynamic and thrust bearings where bearing diameter (mm) $\times$ pump rpm > 300,000 or, pump rated hp $\times$ pump rpm > 2.7×10^6	• Ball bearings with two-year life
Baseplates	• Drip lip or drain pan required • Adequate to limit shaft displacement at coupling to 0.005 in.	• No requirements
Testing	• Hydrostatic test to 1.5 times allowable casing pressure for 30 minutes • Performance test required	• Hydrostatic test to 1.5 times allowable casing pressure for 10 minutes • No performance test required

Applications

Flow conditions: Low head, moderate flow rate

Service rating: Non-critical

Typical uses:
1. Service water
2. LACT charge pump
3. Oil transfer

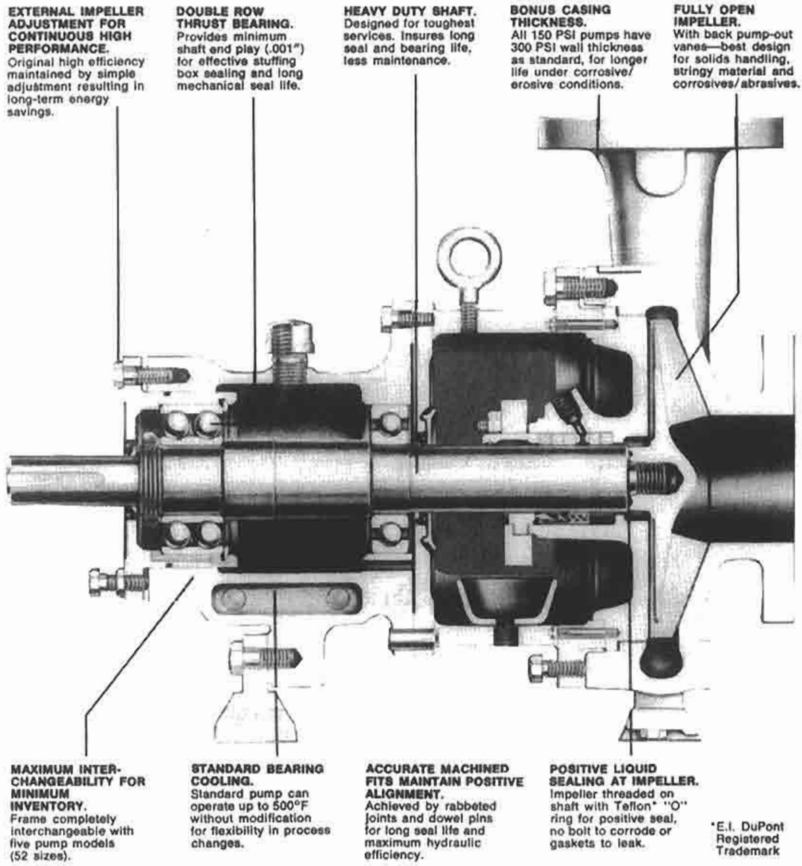


Figure 11-5. ANSI pump (*courtesy of Goulds Pump, Inc.*).

Single-Stage API Pump (Figure 11-6)

Description

Mounting = Horizontal

Casing split = Radial

Impeller type = Radial

Mounting feet = Centerline of pump casing

No. of stages = Single

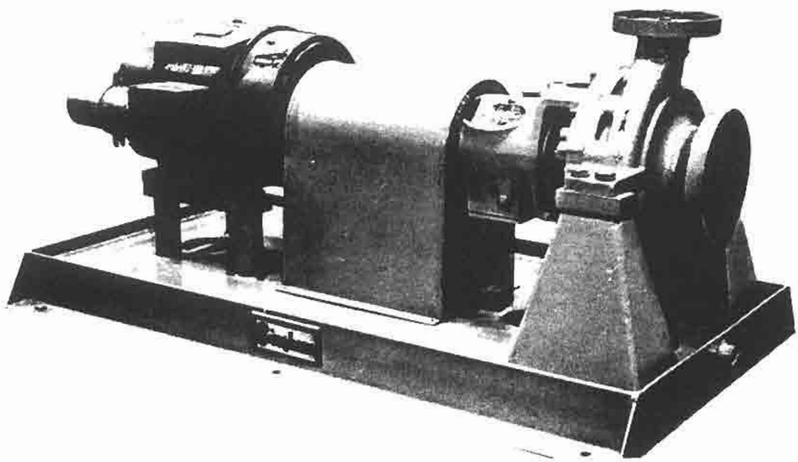


Figure 11-6. Single-stage API pump (*courtesy of Bingham-Williamette Pumps, Inc.*).

Applications

Flow conditions:	Low head, moderate flow rate, high temperature
Service rating:	Critical
Typical uses:	1. Hot oil pumps
	2. Rich oil pumps
	3. Raw products pumps

Vertical In-Line Pump (Figure 11-7)

Description

Mounting = Vertical, in-line with piping
Casing split = Radial
Impeller type = Radial
Mounting feet = None
No. of stages = Single

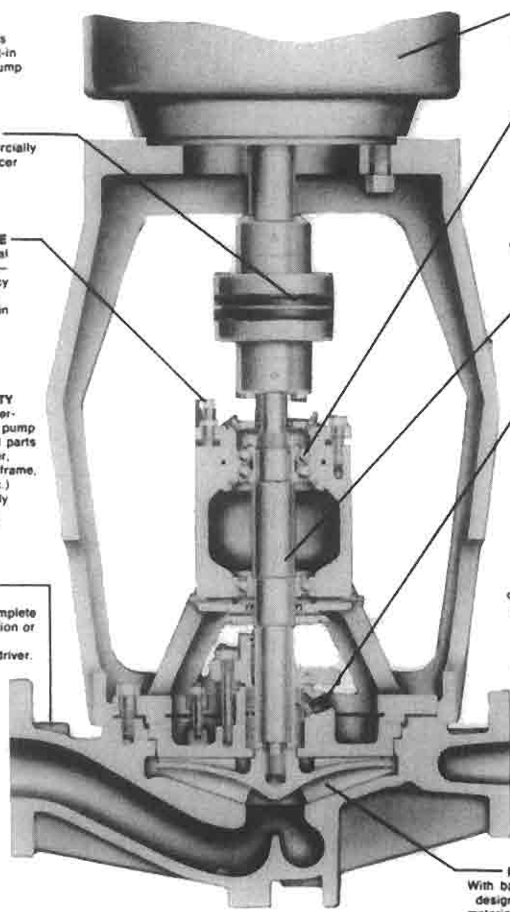


Figure 11-7. Vertical in-line pump (*courtesy of Goulds Pump, Inc.*).

Applications

Flow conditions:	Low head, moderate flow rate
Service rating:	Non-critical
Typical uses:	1. Service water
	2. LACT charge pump
	3. Oil transfer

API Multistage Split Case Pump (Figure 11-8)

Description

Mounting = Horizontal
Casing split = Axial
Impeller type = Radial
Mounting feet = Centerline of pump casing
No. of stages = Multiple

Applications

Flow conditions:	High head and flow rate, high specific gravity
Service rating:	Critical
Typical uses:	1. Pipeline booster
	2. Water flood
	3. Hot oil pumps

API Barrel Pump (Figure 11-9)

Description

Mounting = Horizontal
Casing split = Radial

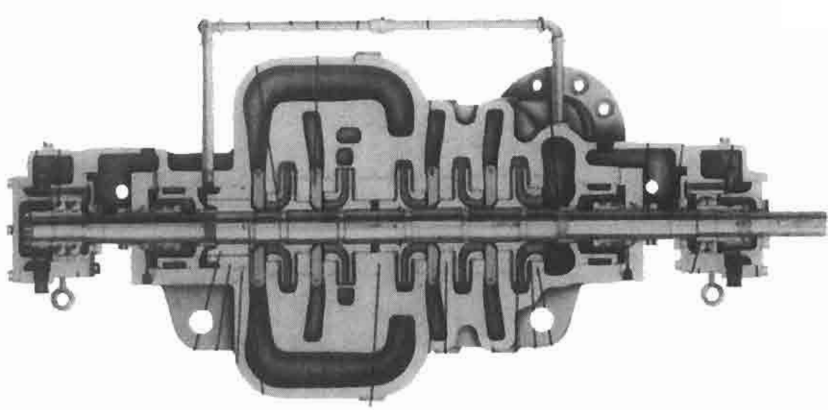


Figure 11-8. API multistage split case pump.

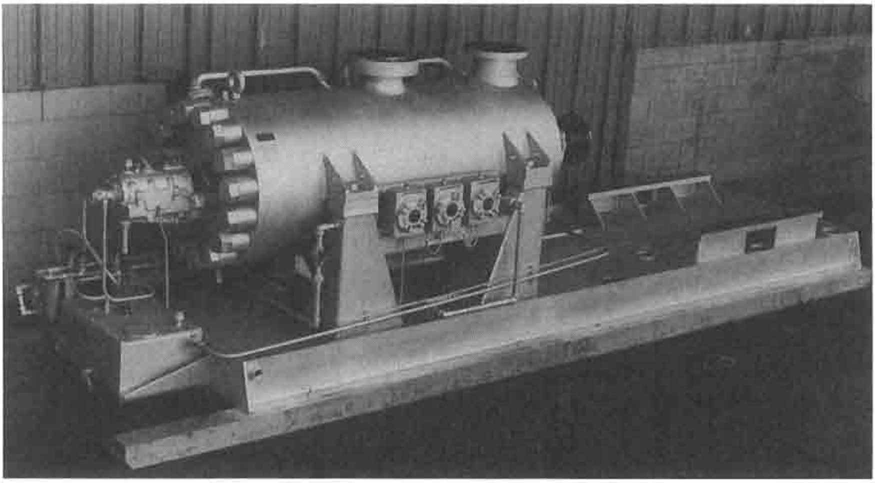


Figure 11-9. API—barrel pump (*courtesy of Bingham-Williamette Pumps, Inc.*).

Impeller type = Radial
Mounting feet = Centerline of pump casing
No. of stages = Multiple

Applications

Flow conditions:	High head and flow rate, all specific gravities
Service rating:	Critical
Typical uses:	1. Pipeline booster
	2. Water flood
	3. Lean oil pumps

Sump Pump (Figure 11-10)

Description

Mounting = Vertical
Casing split = Radial
Impeller type = Radial
Mounting feet = Mounting plate
No. of stages = Single

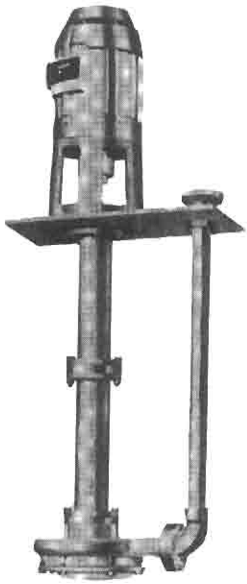


Figure 11-10. Sump pump (*courtesy of Afton Pumps, Inc.*).

Applications

Flow conditions:	Low head and moderate flow rate
Service rating:	Non-critical
Typical uses:	Primarily used to pump water or hydrocarbons from shallow pits or compartments

API Vertical Turbine or Can Pump (Figure 11-11)

Description

Mounting	= Vertical
Casing split	= Radial
Impeller type	= Mixed or radial
Mounting feet	= Mounting plate
No. of stages	= Multiple

Applications

Flow conditions:	High head and high flow rate, low NPSH available

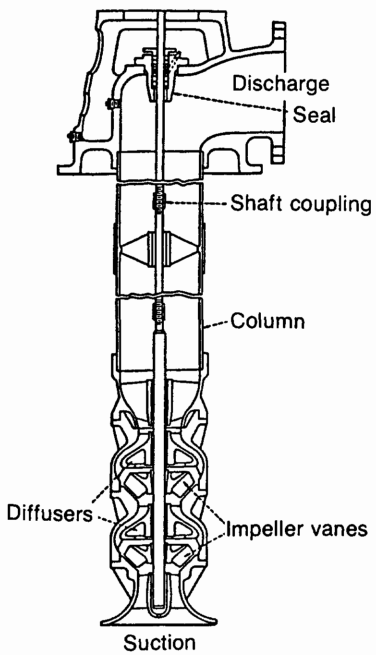


Figure 11-11. API vertical turbine or can pump (*courtesy of Hydraulic Institute*).

Service rating: Critical

Typical uses: 1. Firewater pump offshore
2. Pipeline pump

Submersible Pump (Figure 11-12)

Description

Mounting = Vertical
Casing split = Radial
Impeller type = Radial
Mounting feet = Mounting plate
No. of stages = Multiple

Applications

Flow conditions: High head and moderate flow rate, low NPSH available

Service rating: Critical

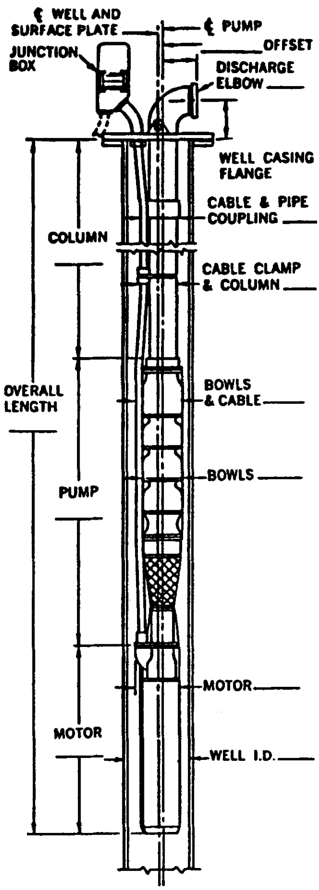


Figure 11-12. Submersible pump (*courtesy of Bryan Jackson, Inc.*).

Typical uses: 1. Firewater pump offshore
2. Downhole in well

BEARINGS, SEALS, AND WEAR RINGS

Bearings

Ball bearings transmit a load from a rotating surface to a fixed surface via a series of rotating balls. They are generally inexpensive, do not require any separate lubrication system, and can be designed to be self-

aligning. However, they have limited ability to handle thrust, high loads, or high speeds.

Roller bearings transmit load from a rotating surface to a fixed surface via a series of rotating cylinders. They have high thrust capability, and do not require any separate lubrication system. However, like ball bearings they have a limited load/speed capability.

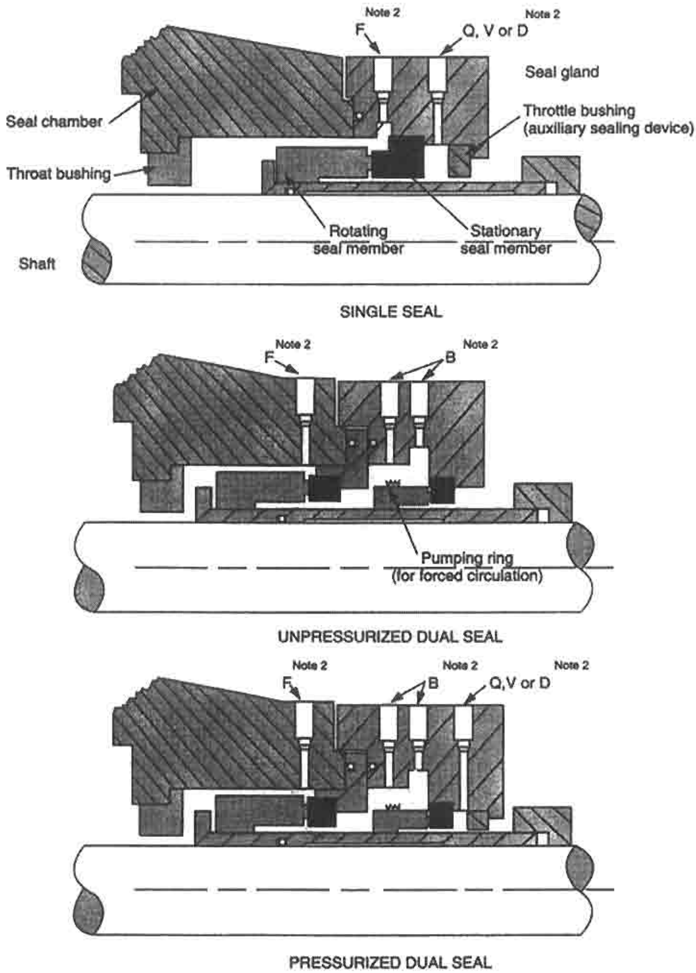
Hydrodynamic sleeve bearings (journal bearings) transmit the load through a thin oil film between a rotating shaft and a fixed bearing surface. They are particularly well suited for high load/speed applications. API 610 requires them on all barrel pumps when the product of the bearing diameter in millimeters times the pump speed in rpm exceeds 500,000 or when the product of pump rated horsepower and pump rpm exceeds 5.4 million. Sleeve bearings have long lives because there is no rubbing between surfaces and they provide some degree of vibration dampening. However, they require a continuous, contaminant-free oil system and have no thrust capability.

Hydrodynamic thrust, or Kingsbury bearings, are the most expensive. The load is transmitted through a thin oil film between a rotating shaft and a fixed bearing surface that consists of multiple pads that tilt. These bearings have all the advantages of sleeve bearings, but in addition they have high thrust loading capability.

Seals

Seals are necessary to prevent leakage at the point where the shaft enters the pump case. The most inexpensive seal is made of packing material that acts as a pressure breakdown device. Packing must be flexible and capable of being compressed for proper operation. Packing material may be made of flexible metallic strands with graphite or oil lubricant impregnation. Asbestos is another material that was common in the past, and though its use has decreased, there are applications where asbestos is the only choice, especially in very-high-temperature service. Besides low cost, packing has the advantage of being easy to replace. However, it has the disadvantages of requiring both some small leakage rate for proper operation, and continuous adjustment as it wears. Packing is generally limited to water service where leakage can be tolerated.

Mechanical seals provide a continuous contact between two flat sealing surfaces located on a plane perpendicular to the shaft centerline as shown in Figure 11-13. They have the advantage of essentially eliminat-



Notes:
1. These illustrations are typical and do not represent any specific design or brand of seal.
2. Refer to appropriate piping plan (see API 610) for required connections.

Figure 11-13. General mechanical seal arrangement (*courtesy of API*).

ing the leakage rate under normal conditions, requiring less frequent adjustment and maintenance, and no run-in time. Besides higher initial and operating costs than packing, they have the disadvantage that when failure occurs it tends to be swift and with large leakage.

There are several types of mechanical seal arrangements. Unbalanced seals are used for pressures less than 150 psi. Balanced seals are used for pressures in excess of 50 psi.

A throttle bushing is a restrictive bushing or sealing device designed to limit flow out of the seal in the event of failure. Since a leaking seal will increase pressure between the seal and throttle bushing, throttle bushings are used whenever a seal failure alarm is required.

Tandem seals are used in critical service where leakage due to seal failure must be prohibited. They are constructed of two seal assemblies acting in series and separated by a buffer fluid at less pressure than the sealing pressure. Should the primary seal fail, the pressure or reservoir level in the buffer fluid system would increase, triggering an alarm. There is normally a throttle bushing and alarm downstream of the secondary seal to provide warning of secondary seal failure as well.

Double seals are used in toxic services where a pressurized clean seal fluid is designed to leak into the lower pressure process should there be a failure in the primary seal. A throttle bushing and alarm downstream of the seal between the clean fluid and the atmosphere is normally installed to warn of failure of this seal.

In order to clarify seal type descriptions with a concise and brief technique an API Seal Classification Code is used. It is a five-letter code described by:

First Letter	B = Balanced
	U = Unbalanced
Second Letter	S = Single
	D = Pressurized dual (double)
	T = Unpressurized dual (tandem)
Third Letter	P = Plain end plate seal gland
	T = Throttle bushing seal gland
Fourth Letter	Gasket material, see Table 11-2
Fifth Letter	Face material, see Table 11-3

Table 11-2
Fourth Letter of Mechanical Seal Classification Code

Fourth Letter	Stationary Seal Ring Gasket	Seal Ring to Sleeve Gasket
E	FKM	PTFE
F	FKM	FKM
G	PTFE	PTFE
H	Nitrile	Nitrile
I	FFKM	FFKM
R	Graphite Foil	Graphite Foil
X	As Specified	As Specified
Z	Spiral Wound	Graphite Foil

FKM—Fluoroelastomer, such as DuPont Viton
PTFE—Polytetrafluoroethylene, similar to DuPont Teflon
Nitrile—B.F. Goodrich Hycar, Buna N or similar
FFKM—Perfluoroelastomer such as DuPont Kalrez

Table 11-3
Fifth Letter of Mechanical Seal Classification Code

Fifth Letter	Sealing Ring Face Material	
	Ring 1	Ring 2
L	Carbon	Tungsten Carbide 1
M	Carbon	Tungsten Carbide 2
N	Carbon	Silicon Carbide
O	Tungsten Carbide 2	Silicon Carbide
P	Silicon Carbide	
X	As Specified	

For example, a BTPFL seal is a balanced, tandem seal with plain end plate, fluoroelastomer "O" rings, a carbon steel seal ring, and a tungsten carbide-1 mating seal ring.

API 610 establishes standard piping systems for mechanical seals. These systems provide a flushing fluid across seal faces, establish flow paths for various seal configurations and establish location for components (e.g., coolers, reservoirs, pressure switches).

Table 11-4 is a general guide to seal usage to provide a starting point for making a cost-benefit decision for any installation.

Wear Rings

These are easily renewable leakage seals between the impeller or shaft and casing designed to prevent leakage from high pressure to low pres-

Table 11-4
Seal Usage Guide

Service	Pressure psi	Seal System	Seal Piping Plans
Water	<100	Packing	
	>100	Balanced mechanical seal	Plan 12 or 13
Hydrocarbons—lease facilities	<100	Mechanical seal	Plan 12 or 13
	100 to 500	Balanced mechanical seal	Plan 12 or 13
	>500 psi	Balanced tandem seal	Plan 12 or 13 with 52
Hydrocarbons—major installations and plants	<100	Mechanical seal with throttle bushing	Plan 12 or 13
	>100	Balanced tandem seals with throttle bushing	Plan 12 or 13 with 52
Toxic fluids	All	Double balanced seal with throttle bushing	Plan 12 or 13 with 52

sure areas. They require a difference in hardness to avoid galling, and in sandy service a flushing fluid for cleaning.

INSTALLATION CONSIDERATIONS

Figure 11-14 shows a typical mechanical flowsheet for two centrifugal pumps installed in parallel. It is presented to illustrate some of the considerations necessary for a good pump installation. Every installation has different objectives and very few will be exactly like this one.

The suction piping is sized for about 2–3 ft/sec and the discharge piping for about 5–6 ft/sec. Although suction and discharge velocities are not as critical for centrifugal pumps as for reciprocating pumps, field experience indicates lower maintenance when the velocities are kept below this range.

Each pump has isolation valves to enable it to be maintained while the other is running. Because of the possibility that a discharge valve could be left open while a suction valve is closed, the suction line up to and includ-

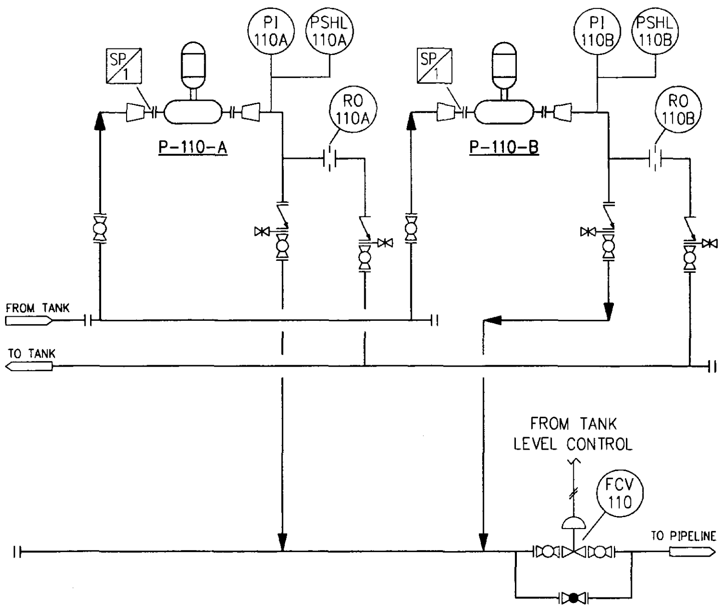


Figure 11-14. Typical mechanical flowsheet for two centrifugal pumps installed in parallel.

ing the block valve should be rated for discharge pressure. As an alternative, a relief valve could be installed in each pump's suction piping.

A check valve is installed in each pump's discharge to prevent reverse rotation of the pump when it is not operating and its isolation valves are open.

A throttling valve (FCV) is installed to control flow without having to start and stop the pumps. It is also possible to start and stop one or both pumps based on level in the feed tank or to use a variable speed driver to control pump speed on level. Since it is possible for the throttling valve to close or for a pump to be started with a blocked discharge, a minimum flow bypass with an orifice (RO) sized to provide sufficient flow to avoid overheating the pump is installed. The bypass is piped back to the tank for further cooling, but this requires a valve on the bypass line that could be left closed, and it does not protect the pump from a closed suction valve. It would be possible to protect against these problems by piping the recycle directly back to the pump suction. The drawback of this type of bypass is that with the short loop from discharge to suction, the liquid will eventually overheat. The continuous bypass effectively reduces the

efficiency of the pump. On large installations it could be attractive to install a pressure control valve on the pump discharge that would only bypass liquid when the pump discharge pressure approached the shut-in pressure. There is much debate on which scheme to use.

It is common to install low- and high-pressure switches in the discharge line (PSHL). The low-pressure switch usually has a delay for start-up. If the pump starts against a closed valve, the pressure will reach the high setting, and the switch will shut the pump down and trigger an alarm that indicates the reason for the shutdown. If the suction valve is closed, the pump pressure will remain low until the low-pressure switch delay expires, and the pump will be shut down. If a large leak develops in the piping downstream of the pump and the pressure drops below the normal operating range, the low-pressure switch will sense the abnormal pressure and shut-in the pump.

The pump is provided with vent and drain connections and a cone-type strainer (SP-1) for startup. Piping must be arranged to allow removal and replacement of the strainer and to allow easy access to the pump for maintenance.

CHAPTER

12

Reciprocating Pumps*

INTRODUCTION

This chapter presents more information on the specification details for reciprocating pumps than was included in Chapter 10. There is no need to discuss head-flow-rate curves for reciprocating pumps, since the flow-rate is merely a function of pump speed and is not dependent on head. Reciprocating pumps operate independently of the system curve (as long as they are mechanically able to meet system pressure). Thus two identical reciprocating pumps in parallel will pump twice the flow of one pump.

First, we will discuss design concepts for minimizing vibrations caused by pulsating flow, then we will discuss specifics about bearings, valve and packing selection, and some of the major provisions of the API standard for reciprocating pumps. Finally we will discuss piping hookup details.

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*Reviewed for the 1998 edition by Jim Cullen of Paragon Engineering Services, Inc.

CONTROLLING PULSATING FLOW

Suction and Discharge Piping

It is absolutely imperative that the suction piping be sized to assure that the NPSH available exceeds the NPSH required by the pump. Often, this can be arranged by elevating the suction tank or by providing a low-head centrifugal charge pump to feed the reciprocating pump. If the NPSH available is too low, valve breakage and pump maintenance costs will be excessive.

If two or more pumps are installed in parallel it is best to install separation suction lines between the tank and the individual pumps. However, in most cases this is not practical and the suction lines are manifolded together. If this is done, the lines should be sized so that the velocity in the common feed line is approximately equal to the velocities in the lateral lines feeding the individual pumps. This avoids abrupt velocity changes and minimizes acceleration head effects.

The suction and discharge piping should be short with the minimum number of elbows and fittings. Where possible, pipe should be laid out using 45° ells for elevation and plan changes rather than 90° ells. Pipe diameter changes should be made with eccentric reducers with the flat side *up* to eliminate gas pockets.

Table 12-1 lists some suggested maximum flow velocities for sizing suction and discharge piping for reciprocating pumps. A low flow velocity for the suction piping is particularly important. Some operators tend to use maximum velocities of one foot per second no matter what the pump speed.

Table 12-1
Maximum Suction and Discharge Pipe Velocities for Reciprocating Pumps

Pump Speed rpm	Suction Velocity ft/sec	Discharge Velocity ft/sec
< 250	2	6
250–330	1.5	4.5
> 330	1	3

Pulsation Dampeners

In many instances where the sizing recommendations previously discussed are used, there is not need to install dampeners to reduce liquid

pulsations. However, in most instances, it is possible to further reduce pulsations and thus reduce pump maintenance costs and piping vibration with the use of pulsation dampeners. Dampeners are recommended for all major multipump installations, unless computer analog studies indicate that they are not needed. In many instances it is cheaper to install the dampeners than to perform the detailed engineering studies to prove that they are not needed. Dampeners can be either liquid-filled, gas-cushioned, or tuned acoustically as shown in Figure 12-1.

A liquid-filled dampener is merely a large surge vessel located close to the pump. It uses the compressibility of the liquid itself to absorb pressure pulsations and thus works better on gaseous liquids (e.g., hydrocarbons, rich glycol) than on relatively gas-free liquids. The volume of the vessel is normally recommended to be ten times the pump displacement (volume per minute). Thus, for single acting pumps:

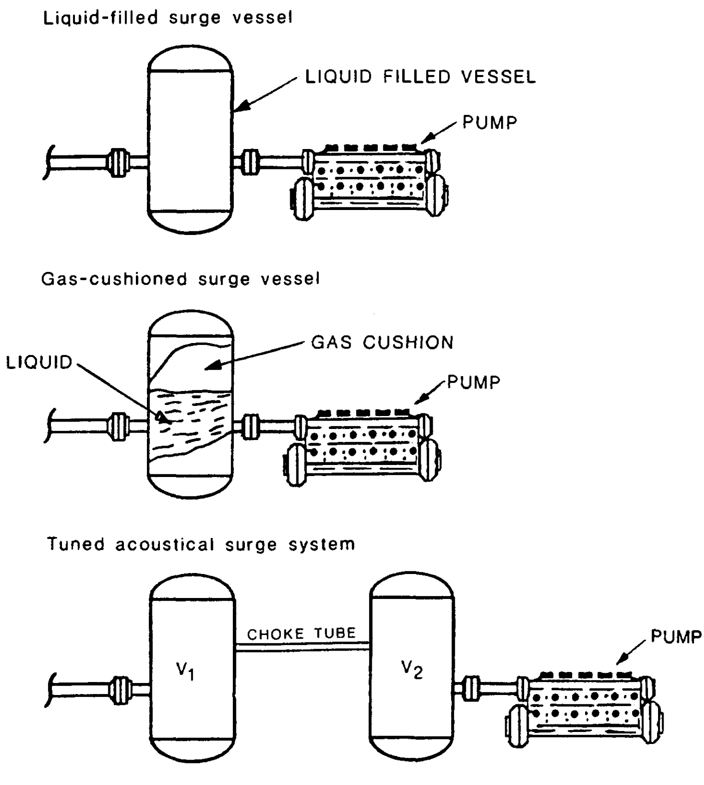


Figure 12-1. Typical pulsation dampeners.

$$Vol = 0.04\ Q_1 \tag{12-1}$$

where Vol = volume of surge tank, ft^3
Q_1 = flow rate, bpd

Derivation of Equation 12-1

Vol is in ft^3, Q_1 in bpd, Q in ft^3/sec

$$Vol = (10)(Displacement)$$

$$Displacement = (60)\ Q$$

$$Q = Q_1 \times \frac{(5.61)}{(24)(3{,}600)} = 0.65 \times 10^{-4}\ Q_1$$

$$Vol = 0.04\ Q_1$$

Liquid-filled surge vessels are maintenance-free and have a negligible pressure drop associated with them. However, they tend to take up a large amount of space and are heavy due to the weight of liquid. Since the vessels must be rated for the same pressure as the pipes to which they are attached, they are expensive. Figure 12-2 shows an installation for a quintuplex pump with liquid surge suction and discharge dampeners.

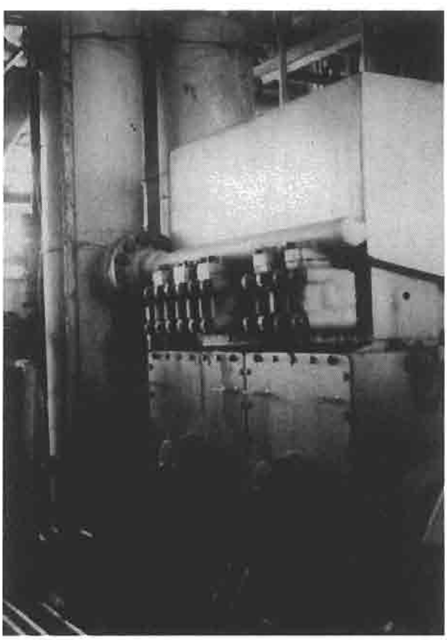


Figure 12-2. Quintuplex pump with liquid surge suction and discharge dampeners.

Typical gas-cushioned dampeners are shown in Figure 12-3. The simplest type is merely a surge bottle with a gas-liquid interface. The high compressibility of the gas provides absorption of the pressure pulses. The gas in the vapor space is normally natural gas, which over a period of time can dissolve in the liquid. Conversely, it is also possible for gas to flash from the liquid being pumped. Thus, a level gauge is installed so that the interface position can be observed and either more gas added or excess gas vented to maintain the required gas volume.

The required gas volume is given by:

$$(\mathrm{Vol})_g = \frac{Ksd^2P}{100\,(\Delta P)} \tag{12-2}$$

where $(\mathrm{Vol})_g$ = required gas volume, gal
K = pump constant from Table 12-2
s = pump stroke, in.
d = pump piston or plunger diameter, in.
P = average fluid pressure, psi
ΔP = allowed pressure pulsation, psi

Table 12-2
Constant for Gas-Cushion Design

Pump Type	Action	K
Simplex	Single	0.67
Simplex	Double	0.55
Duplex	Single	0.55
Duplex	Double	0.196
Triplex	Single	0.098
Triplex	Double	0.196
Quintuplex	Single	0.030
Quintuplex	Double	0.060
Septuplex	Single	0.017
Septuplex	Double	0.034

The allowable pressure pulsation amplitude is somewhat arbitrary. Figure 12-4 can be used as an estimate if other information is not readily available.

Gas-cushioned dampeners are much smaller than liquid-filled dampeners but require monitoring of the interface. They have the drawback of not being practical in locations where the discharge pressure varies widely and the gas volume expands and contracts in response.

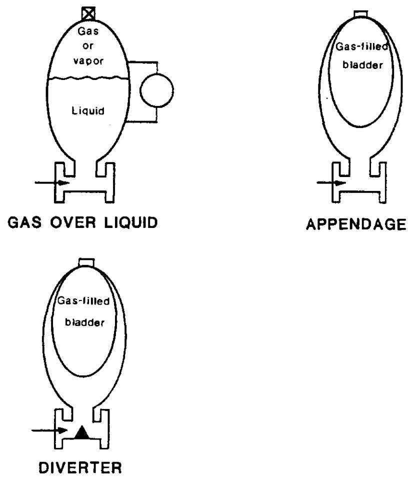


Figure 12-3. Typical gas-cushioned dampeners.

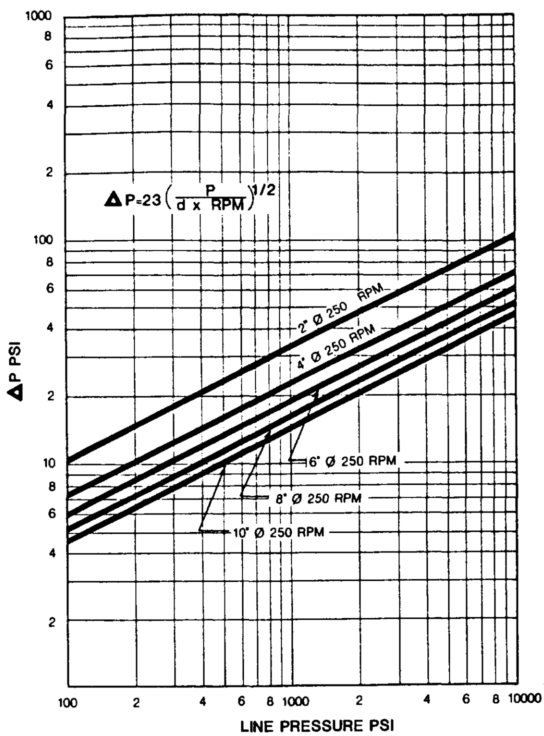


Figure 12-4. Approximate allowable pressure pulsations.

Most gas-cushioned dampeners employ a pressurized bladder to keep the gas from being absorbed in the liquid. They can have a configuration such as that shown in Figure 12-3, or the bladder can be in the shape of a cylinder such as the in-line bladder of Figure 12-5. The use of diverters in appendage-type dampeners or in-line configurations aids in attenuating high-frequency pulsations.

The size of the gas volume depends upon the bladder properties and configuration of the design. For approximating purposes, Equation 12-2 can be rewritten:

$$(\text{Vol})_g = \frac{\text{Ksd}^2\text{P}^2}{100\,(\Delta\text{P})\,\text{P}_c} \qquad (12-3)$$

where P_c = bladder precharge pressure, psi

The bladder precharge pressure is normally set at 60–70% of average fluid pressure. Bladder type desurgers have the disadvantage that the elastomer can eventually wear out and need to be replaced, and is limited to applications below about 300°F. However, they are normally an economical solution and are in common use where it is not expected that frequent attention will be paid to the gas-liquid interface.

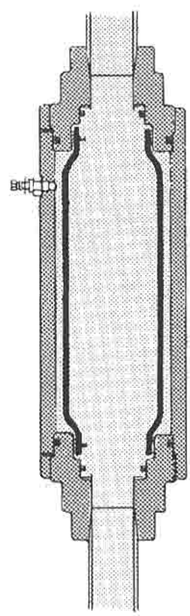


Figure 12-5. In-line desurger.

Tuned acoustical dampeners are formed when two liquid-filled vessels are connected by a short section of small diameter pipe called a choke tube. This system can be designed to have a specific resonant frequency. Pressure pulsations at frequencies above this level are attenuated considerably. They are excellent for high-frequency pulsations, can be used in high-temperature situations, and are essentially maintenance free. However, they have the disadvantages of high cost, they take up a lot of space, and have a relatively high pressure drop through the choke tube when compared to the other alternatives. For this reason they are not normally used on suction lines where NPSH may be a problem. Figure 12-6 shows a pump installation with an appendage bladder dampener on the suction and a tuned acoustical filter on the discharge. The pressure vessel contains two sections, one above the other, with a choke tube connecting them internally.

An extremely efficient type of dampener can be made by connecting two gas cushion dampeners in series with a short run of pipe that acts as a choke tube. The gas-cushion dampeners attenuate low-frequency pulsations and the choke tube arrangement serves to alleviate high-frequency pulsations. Most installations do not require this complexity.

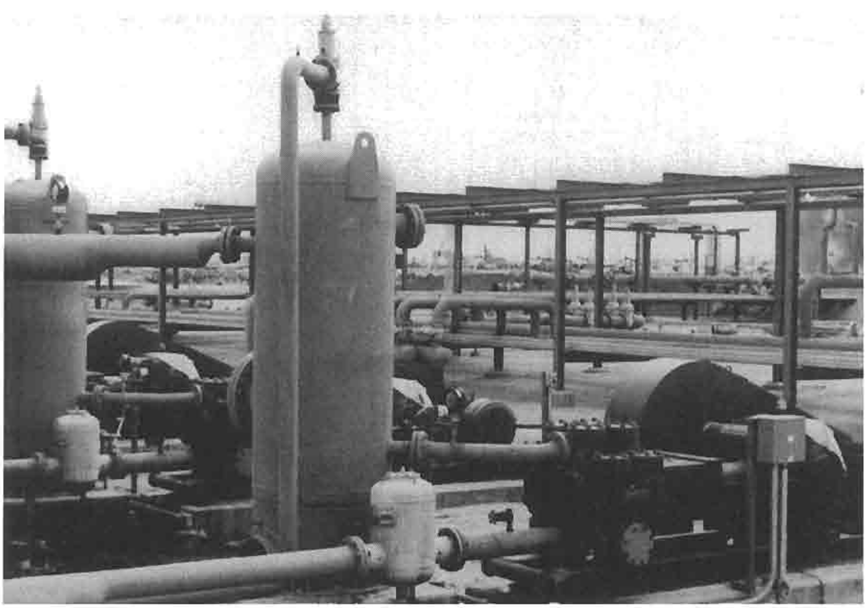


Figure 12-6. Pump with an appendage bladder dampener on the suction and a tuned acoustical filter on the discharge.

The design of acoustical filters is best done on computer analogs and is beyond the scope of this text.

Pipe Vibrations

To minimize pipe vibrations it is necessary to design pipe runs so that the "acoustic length" of the pipe run does not create a standing wave that adds to the pressure pulsations in the system. The acoustic length is the total overall length from endpoint to endpoint including all elbows, bends, and straight pipe runs. Typical pipe runs with respect to acoustic length are considered to be:

1. Pipe length from suction tanks to the pump suction.
2. Long pipe sections between pump and pulsation dampener.
3. Pipe section between pump and manifold.

Piping should be designed so that piping runs with "similar" ends should not equal 0.5λ, λ, 1.5λ, 2λ, ... where λ is the acoustic wavelength. Pipe runs with "dissimilar" ends should not equal 0.25λ, 0.75λ, 1.25λ, 1.75λ, ...

The ends of pipe runs are considered similar if both are either open or closed from an acoustic standpoint. They are dissimilar if one is open and one is closed. Examples of end classifications are:

1. If pipe size is dramatically reduced, it tends to act as a closed end.
2. Orifice plates act as closed ends.
3. Short-length flow nozzles act as closed ends.
4. Abrupt pipe diameter enlargements act as open ends.
5. Pipe tees presenting an increase in flow area (such as a tee with three equal legs) act as open ends.
6. Pipe size changes that occur smoothly over a pipe length corresponding to several pipe diameters do not act as a termination.

The acoustic wavelength is given by:

$$\lambda = \frac{720\left(\frac{gB_1}{\rho}\right)^{1/2}}{(\text{rpm})\, n} \tag{12-4}$$

where B_1 = bulk modulus of the fluid, from Figure 12-7, psi
$g = 32.2$ ft/sec^2

ρ = fluid density, lb/ft^3
rpm = pump speed
n = number of plungers

It is also desirable to assure that the natural frequency of all pipe spans is higher than the calculated pump pulsation frequency to minimize mechanical pipe vibrations. The pump pulsation frequency is given by:

$$f_p = \left(\frac{rpm}{60}\right)n \qquad (12-5)$$

where f_p = pump pulsation frequency, cycles/sec

The natural frequency of pipe spans can be estimated from Figure 12-8. Normally, if the pipe support spacing is kept short, the pipe is securely tied down, the support spans are not uniform in length, and fluid pulsations have been adequately dampened, mechanical pipe vibrations will not be a problem.

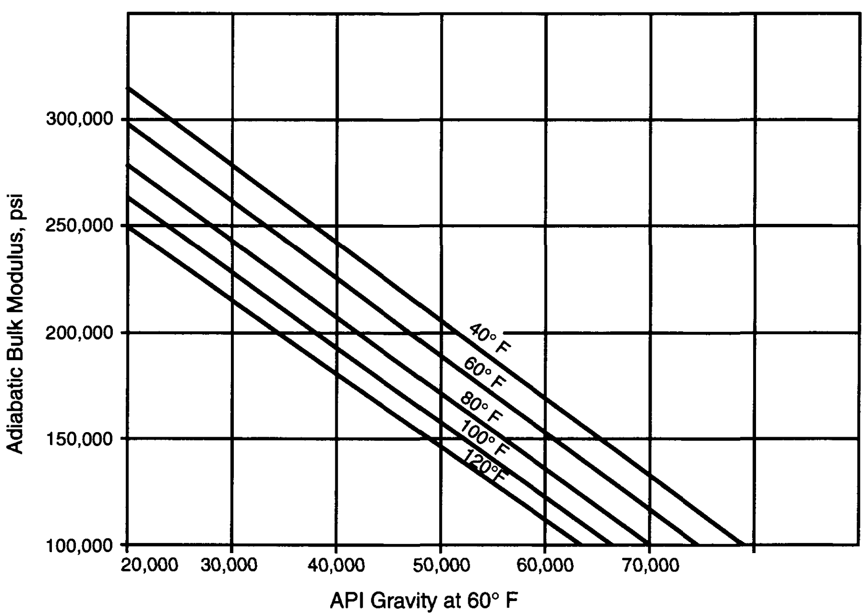


Figure 12-7. Adiabatic bulk modulus vs. API gravity.

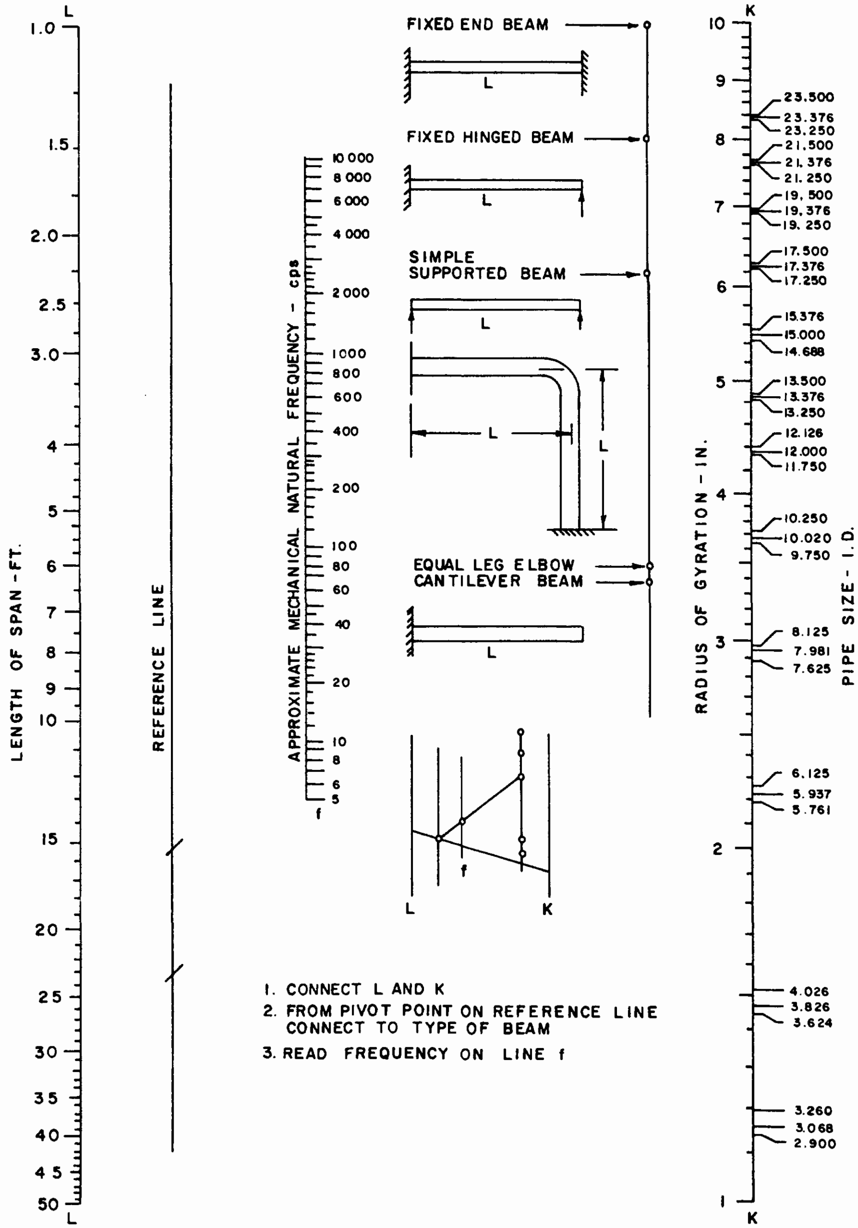


Figure 12-8. Natural frequency of pipe spans (*courtesy of Southwest Research Institute*).

BEARINGS, VALVES, AND PACKING

Bearings

The bearing locations in a typical horizontal plunger pump are shown in Figure 12-9. Bearings can be either ball, roller, or journal depending upon the manufacturer's design. The advantages and disadvantages of each type are discussed in Chapter 11.

Valves

The various types of valves are shown in Figure 12-10. Plate, wing and plug valves are normally limited to clean fluids because of the flow pattern through the open valve. There tends to be a much more immediate increase in flow area when a ball valve first begins to open. Slurry valves have hardened inserts to provide a longer lasting seal.

Packing

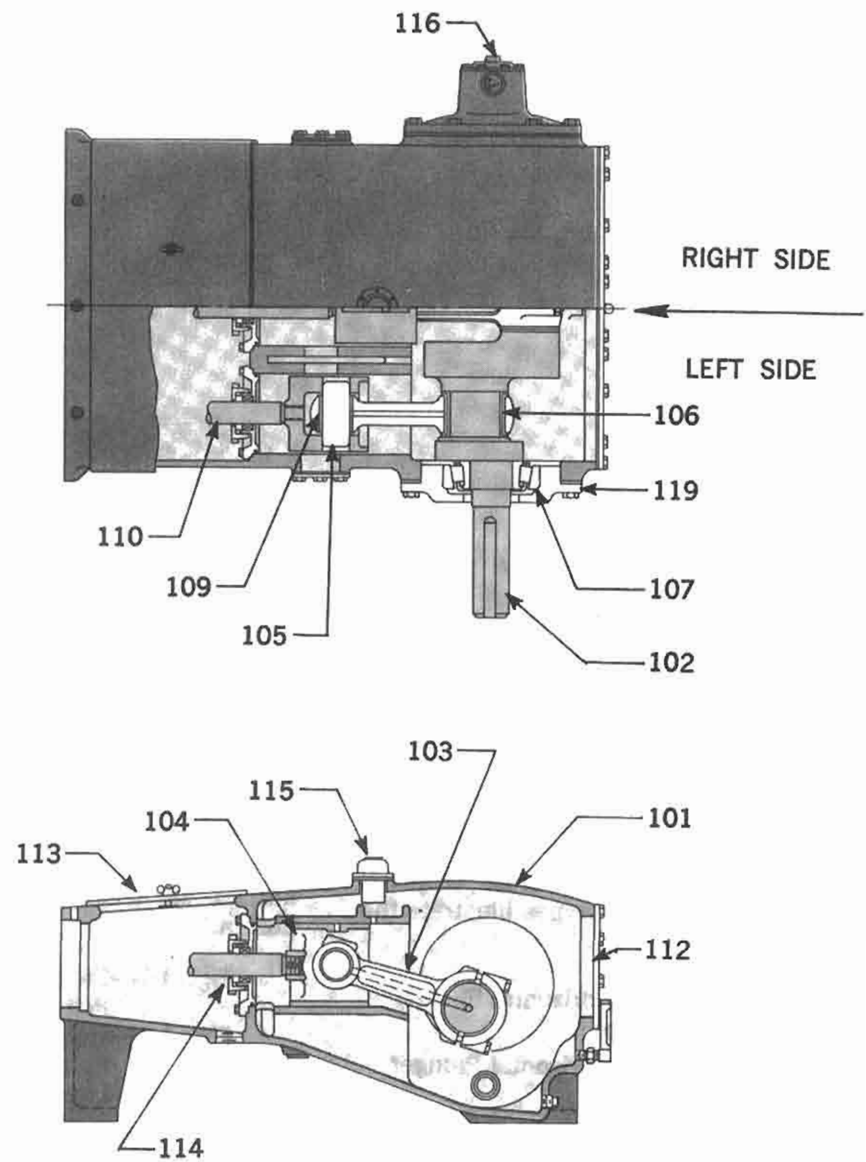
Packing is made from metallic strands with graphite or oil impregnation and is used to seal the location where the plunger or piston rod enters the fluid cylinder. Packing is either square cut or chevron in design as shown in Figure 12-11.

Square-cut packing is less expensive, and easier to repair. However, it requires continuous adjustment of the gland as the packing wears and is not recommended when solids are present in the fluid. Chevron packing is self-adjusting as the packing wears and has less leakage associated with it. The spring provides a cavity for injection of clean seal fluids in dirty service. However, chevron packing can reduce volumetric efficiency for high-compressibility fluids as it is in communication with the cylinder and vapors can accumulate in the cavity.

CODES AND STANDARDS

Most reciprocating pumps in oilfield service are built to manufacturer's standards, although they have many features required by API 674, Positive Displacement Pumps Reciprocating. The basic requirements of API 674 are summarized below to provide some guidance in making selections of various options:

- *Pressure Ratings*—All pressure and temperature ratings encountered in oilfield production operations.



101	Frame	107	Bearing, Main Crankshaft	114	Box, Wiper
102	Crankshaft	109	Bearing, Wrist Pin	115	Breather
103	Rod, Connecting	110	Extension, Crosshead	116	Pump, Lube Oil
104	Crosshead, Power	112	Cover, Crankcase	119	Housing, Bearing, Crankshaft
105	Pin, Wrist	113	Cover, Cradle		
106	Bearing, Crankpin				

Figure 12-9. Bearing locations in a typical horizontal plunger pump (*courtesy of Hydraulic Institute*).

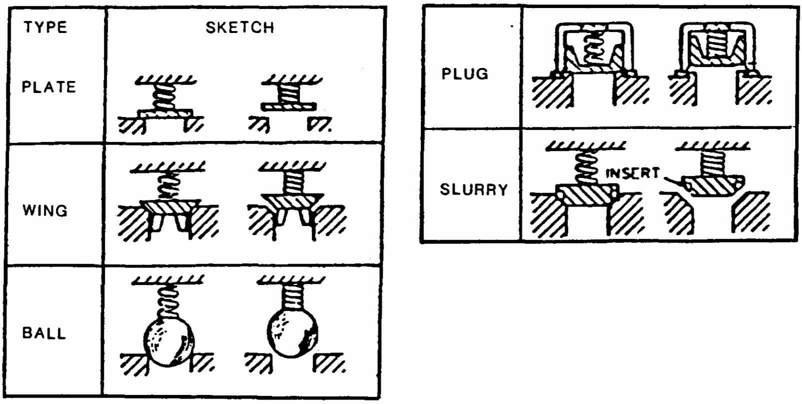


Figure 12-10. Typical valve types.

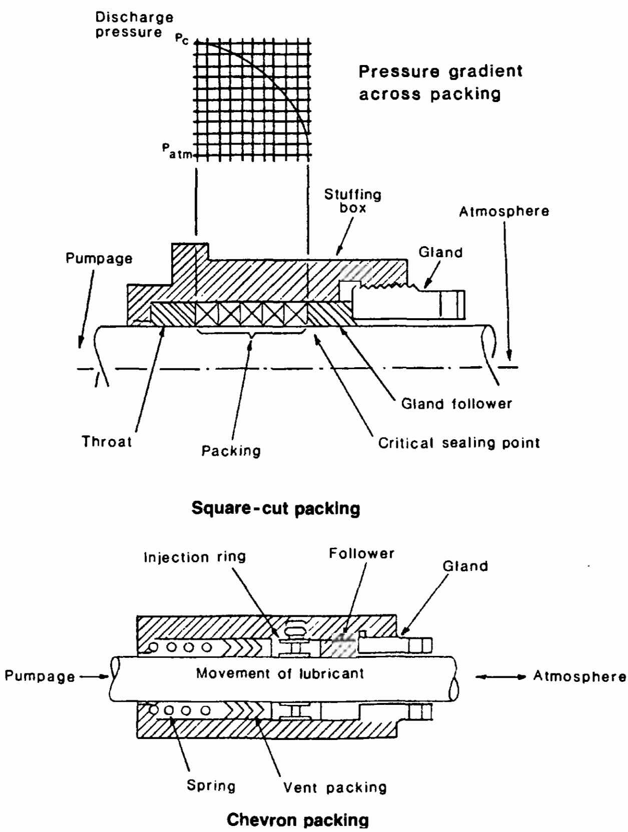


Figure 12-11. General packing arrangements.

- *Pump Speed*—Speeds for single-acting plunger pumps are limited to the values shown in Table 12-3.
- *Cylinder Design*—Requires ASME code Section VII Division 1 for cylinders. Forged cylinders required for applications above 3,000 psi.
- *Cylinder Liner*—Required.
- *Plunger or Rods*—Requires that all piston rods or plungers in contact with packing be hardened or coated to Rockwell C35.
- *Stuffing Boxes*—Required to accept at least three packing rings.
- *Bearings*—Ball bearings must be capable of three years of continuous operation at rated pumping conditions.
- *Materials*—Cast iron or ductile iron is not allowed for pressure-retaining parts handling flammable or toxic fluids.
- *Testing*—Pressure-retaining parts (including auxiliaries) are tested hydrostatically with liquid at a minimum of 1½ times the maximum allowable working pressure.

Table 12-3
API 674
Maximum Allowable Speed Ratings for Power Pumps in Continuous Service

	Speed Rating			
	Single-Acting Plunger-Type Pumps		Double-Acting Piston-Type Pumps	
Stroke Length (inches)	rpm	ft/min	rpm	ft/min
2	450	150	140	46.5
3	400	200	—	—
4	350	233	116	77
5	310	258	—	—
6	270	270	100	100
7	240	280	—	—
8	210	280	—	—
10	—	—	83	138
12	—	—	78	156
14	—	—	74	173
16	—	—	70	186

Note: For an intermediate stroke length, the maximum speed shall be interpolated from the numbers in the table.

PIPING HOOKUP

Figure 12-12 shows a typical hookup for two reciprocating pumps operating in parallel. Because the pump can be accidentally started when the discharge block valve is closed, a relief valve is installed in the discharge line to keep the pump from overpressuring the pipe and flanges. It is also possible to leave the suction valve closed while the discharge valve is opened. Discharge fluid could leak through the discharge check valve and pump valves pressuring up the suction piping which is rated for 150 ANSI. Thus, a relief valve is installed in pump suction piping.

An appendage dampener (SP-3) and cone strainer (SP-1) are installed in the suction. An in-line desurger and check valve are installed in the

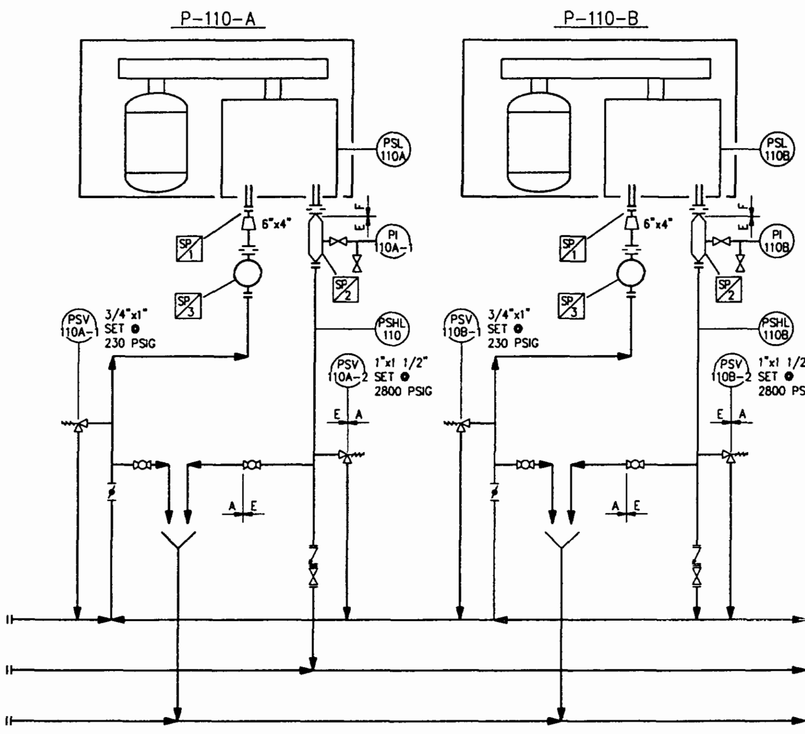


Figure 12-12. Typical mechanical flowsheet for two reciprocating pumps operating in parallel.

discharge. The check valve protects against leakage from discharge when the pump is not running. It is preferable that this be a piston check valve to keep it from chattering due to pressure pulsations. Many successful installations, which follow the design practices to minimize pulsations previously described, use swing check valves.

Drain valves are provided so that the pump can be easily maintained and a low oil-pressure switch (PSL-110A and 110B) is provided to shut in the motor.

API RP 14C, Surface Safety Systems for Offshore Production Platforms, requires that a high-pressure sensor be installed on the discharge so that the pump will shut down before the discharge relief valve opens. It also requires that a low-pressure sensor on the discharge be installed to shut down the pump in case of a large leak in the discharging piping. These two functions are carried out by one device (PSHL-110A and PSHL-110B).

OPERATION

Bypass Valve

In order to reduce the full-load torque required from the driver, an unloader or bypass valve may be installed in the pump discharge. This valve bypasses flow from the discharge back to the suction during start-up; this arrangement has the advantage of reducing start-up torque. The arrangement also results in lower rod loading, allowing better lubrication of sleeve-type bearings at this crucial time when the pump is often cold.

The valve is progressively closed as the pump achieves operating conditions. This procedure may be done manually, or it may be pre-programmed for particular operating circumstances.

Variable-Speed Motors and Drivers

In addition to the normal mechanical drivers available, electric AC motors of the standard squirrel-cage type can be used in combination with a Variable-Frequency Driver (VFD) to allow variable-speed operation of the pump. By varying the speed of a reciprocating pump, one can adjust the flow rate through the pump to meet operational requirements.

The VFD/AC motor combination, which is competitive with mechanical drivers, has the advantages of low maintenance and no emissions. In

addition, no additional fuel or steam systems—which may be required with mechanical drivers—are needed. This advantage can prove useful on some offshore platforms and remote sites.

The main disadvantage on large pumps (e.g., over 500 horsepower) using VFD is that the installation becomes large and expensive. The VFD also produces many harmonics that can be deleterious to other sensitive equipment on the same electric bus. Special harmonic filters and/or isolation transformers are needed to alleviate this problem. In addition, cooling requirements increase due to heat generated by the drive and auxiliary equipment. In spite of these factors, variable-speed AC motors are becoming more popular.

CHAPTER

13

Organizing the Project*

INTRODUCTION

The concept of project management may be defined as the art, act, or manner of managing, handling, controlling, and directing an engineering project. A characteristic of project management is its one-time-through nature. Project management differs from general business management in the time element. Where a business grows and develops over a period of several years, a typical project is usually compressed into a few months or a few years.

Members of a project team are brought together for the duration of the project only and must be quickly organized to accomplish specific objectives. The business manager has the luxury of time to evolve a method unique to her concern for controlling and directing. The project manager, on the other hand, must organize his efforts around more or less standard tools to assure that he and the members of his team can quickly develop lines of communication and control.

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*Reviewed for the 1998 edition by Sandeep Khurana of Paragon Engineering Services, Inc.

PROJECT STEPS

Project Initiation

The beginning of a project occurs at the time objectives or goals are established. This normally includes a brief understanding of the nature and extent of the work, a tentative schedule, and possibly a target cost. Normally, the project will be initiated with the success of one or more exploratory wells or with a geological or reservoir study indicating the need for an expansion of existing production facility capabilities.

Conceptual Study

The conceptual study is the first step of any project. It investigates one or more means of accomplishing the objective. An economic and technical assessment and comparison of the various methods or schemes is made. The owner compares various alternatives by using discounted cash flow analysis. The analysis includes estimations of the timing and amounts of all projected capital and operating expenditures and related revenues. Generally, one determines the financial acceptability of a capital investment by calculating the net present value (NPV) within an acceptable discount rate of return for each alternative. The selected alternative is the one with the greatest NPV. Other discounted cash flow parameters such as internal rate of return, payout period, percent profit, etc., are also used to facilitate choosing among competing projects for limited budget funds.

Figure 13-1 shows the various development schemes that might be analyzed for an offshore field development. The basic tools used in economically assessing these various developmental schemes are block diagrams and analyses of alternative costs.

Block Diagram

Figure 13-2 is a simple block diagram for a field where all the functions are performed at one location. Other alternatives could be investigated that have satellite facilities with some or all the equipment installed at each location. The block diagram develops a selected process or alternative into specific descriptions and recommendations for equipment. Types and arrangements of equipment would be studied, and a design philosophy established.

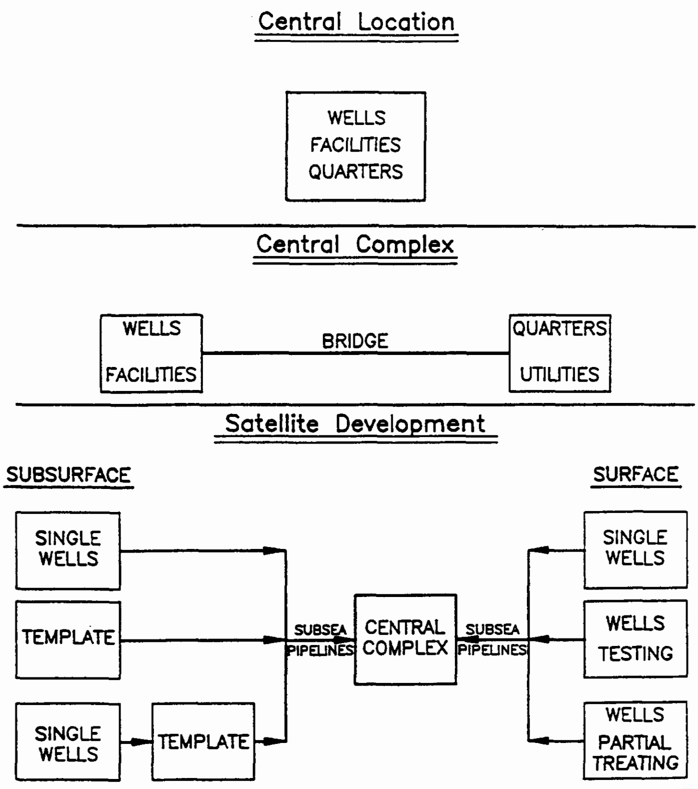


Figure 13-1. Example development schemes.

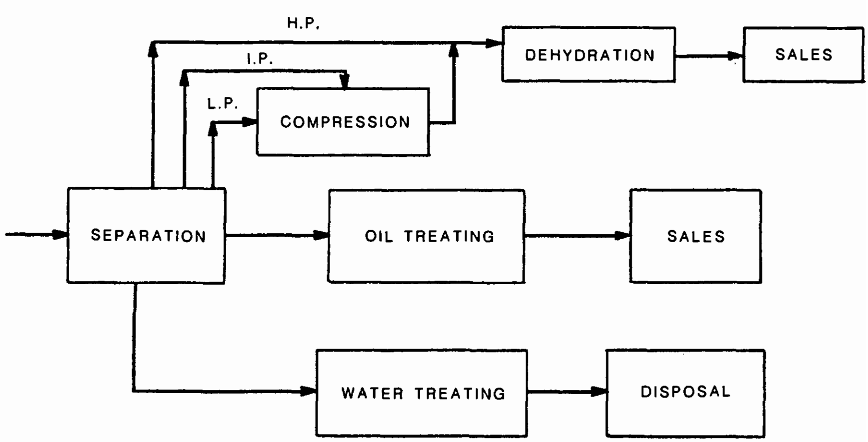


Figure 13-2. Typical oil facility block diagram.

Structural Concept

The purpose of the structure is to support the facilities. Onshore, the cost of preparing the foundation is generally low compared to that of the production facilities. Offshore, however, the cost of the structure—depending on the water depth—can be considerably higher than the cost of the facilities.

A conventional template-type offshore platform such as that shown in Figure 13-3 is suitable in mild environments at water depths of up to 1,300 feet. For shallow water depths (up to 200 feet), several so-called "minimum" concepts are available. These concepts are limited by the amounts of topsides facilities weight that they can support. Minimum concepts use light-weight structures and have structural framing that provides less redundancy than that of a typical four-leg or three-leg welded space frame template-type platform. The light weight of minimum designs results in lower material and fabrication costs. To evaluate whether a minimum concept is economical, however, one must look at the overall cost, including installation and maintenance, and account for the risk of failure, which tends to be higher for non-redundant structures, especially those with bolted connections. In most instances in water depths of more than

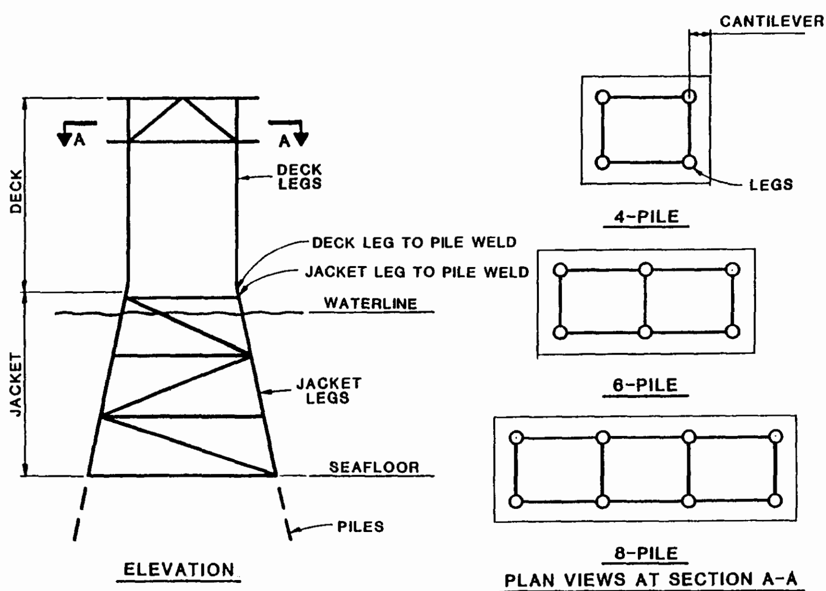


Figure 13-3. Typical offshore platform.

100 feet, when overall cost is considered, welded space frame tripod structures (shown in Figure 13-4) are either less expensive or just marginally more expensive than most other less-reliable structures.

Preliminary Cost Analysis

Simultaneously, an analysis of the cost and economic benefits of each alternative is performed. Such an assessment provides a basis for a logical choice of methods, processes, schedules, etc.

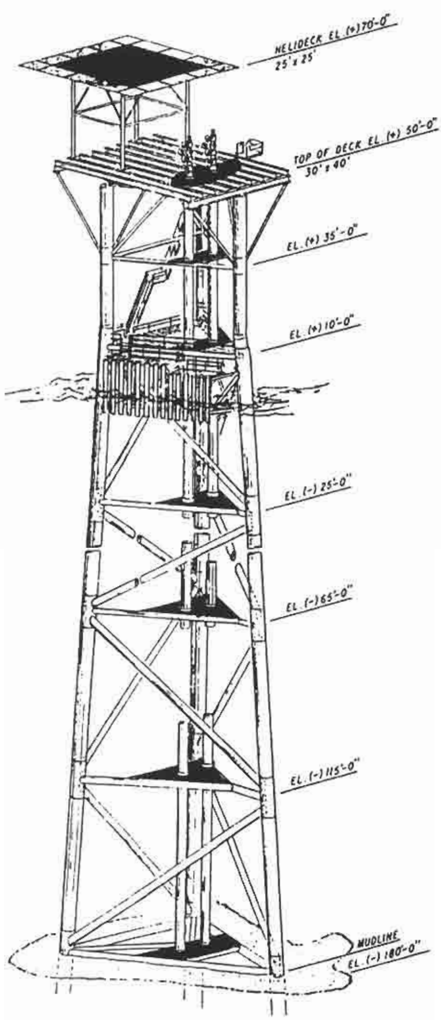


Figure 13-4. Welded space frame tripod by Paragon Engineering Services, Inc.

Figure 13-5 shows a preliminary cost analysis comparing two alternatives. At this stage it may be more important to identify differences in costs between alternative schemes than to determine the absolute cost of the project. For this reason, it is probably wise to use gross estimation techniques rather than detail equipment takeoffs.

	Items	Option I Crude Treating Offshore $M	Option II Crude Treating Onshore $M	Option III Minimum Treating Offshore $M
	OFFSHORE			
A.	Crude Treating Skid	430.0	37.8	332.9
B.	Other Process Equipment	239.5	100.0	144.5
C.	Compressor Package	600.0	600.0	225.0
D.	Generator Station	34.0		14.0
E.	Platform Utilities	50.0		
F.	Quarters Building	200.0		
G.	Instr. and Shutdown System	105.0	35.0	30.0
H.	Offshore Installation	924.0	490.0	546.8
I.	Miscellaneous	810.0	365.0	427.0
J.	Pipeline	677.2		677.2
	ONSHORE			
A.	Site Preparation		350.0	
B.	Production Skid		180.2	
C.	Other Process Equipment		142.0	
D.	Onshore Installation		175.0	
E.	Miscellaneous		80.0	
Subtotal $M		4069.7	2555.0	2397.4
15% Contingency		610.5	383.3	359.6
Total $M		4680.0	2940.0	2760.0

Figure 13-5. Example preliminary cost analysis.

Project Definition

The next phase in project management for the scheme selected in the conceptual study is to define the project. The tools used to define the project are process flowsheets, layout drawings, preliminary cost estimates, and project execution plans.

Process Flowsheet

The block diagram is converted into a process flowsheet so as to better define the project. The process flowsheet shows all major equipment. It includes main piping with flow arrows, and shows operation pressure and temperature of the piping and equipment. The major instrumentation that controls the main process flow is shown and every major line is assigned a stream number. A table is included listing pertinent design data for these streams. Data to be listed include flow rates, pressures, temperatures, specific gravities, and other properties when required. Figure 13-6 illustrates a typical process flowsheet.

A process flowsheet does not show utilities, safety systems, firewater systems, spare equipment, minor or support lines, detailed instrumentation or equipment item numbers, valves, etc.

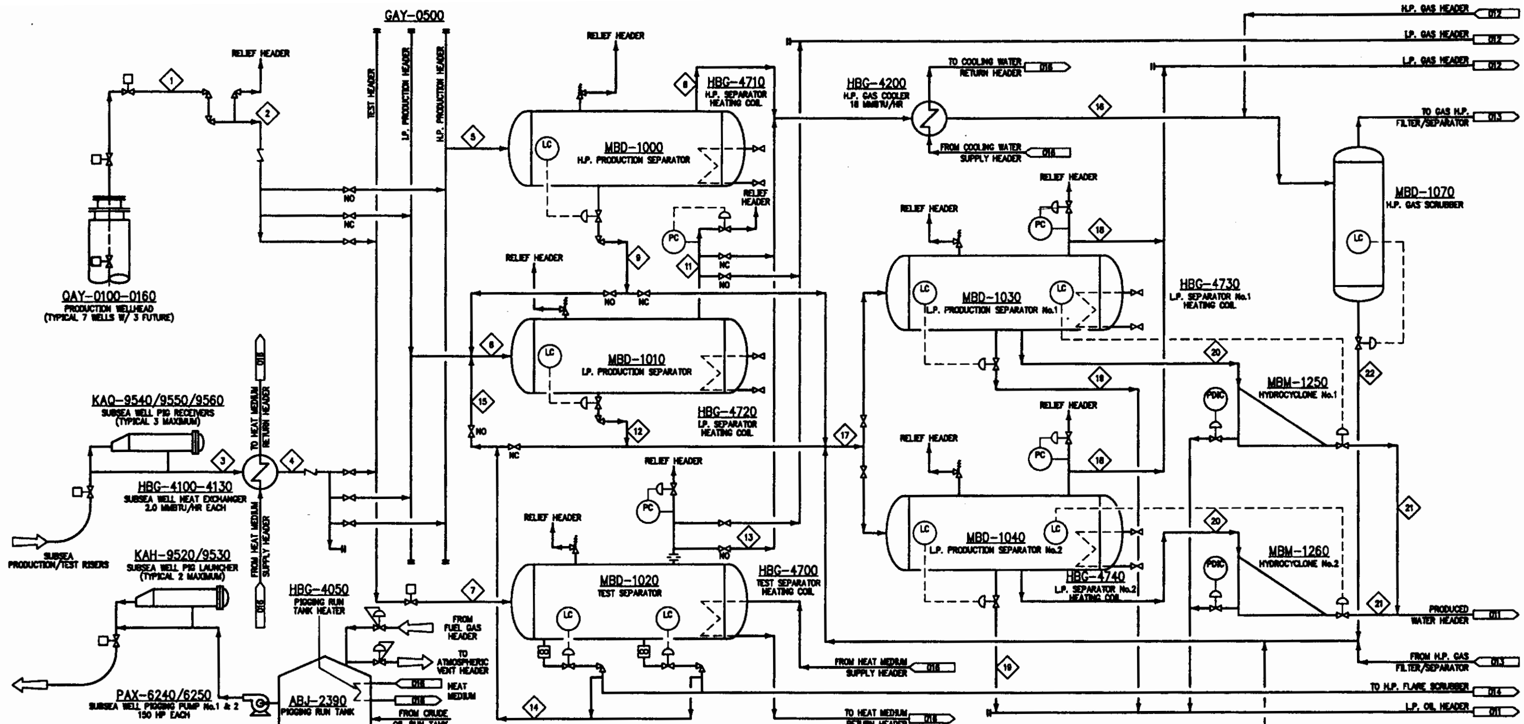
Layout Drawings

The layout drawing locates the equipment defined on the process flowsheets. A well planned layout is the key to good operation, economical construction, and efficient maintenance. The layout drawing must be integrated with the development of the process flowsheet and must be settled before detailed piping, structural, and electrical design can be started. Many times the process can be simplified by the judicious layout of atmospheric vessels. This is particularly true for drain and water treating equipment.

Good layout planning is usually the result of experienced judgment rather than prescribed rules and for this reason should be a joint effort of the most experienced engineers. This is particularly true of offshore platforms where space is very expensive. A large part of layout design involves separating pieces of equipment that could present a hazard to each other, assuring clear escape routes and allowing for machinery maintenance. Too often an inexperienced engineer will concentrate on minimizing piping and neglect these concepts. Figure 13-7 is a typical layout for the production skid portion of an offshore platform.

Preliminary Cost Estimate

The preliminary cost estimate is an important tool in the generation of the initial authority for expenditure (AFE). For effective cost control the preliminary cost estimate must be made accurately and upgraded when



STREAM NO.	1	2	3	4	5	6	7	8	9	10	11
DESCRIPTION – 60,000 BPD CASE											
TEMPERATURE (°F)	140	150	100	130	150	150	150	150	150		150
PRESSURE (PSIG)	7000	1900	1900	1900	1900	625	1900	1900	625		625
GAS FLOW (MMSCFD)	136.3	167.8	50	50.9	141.6	15.8	26	141.8	13.4		15.8
GAS S.G.	0.67	0.65	0.63	0.64	0.65	0.67	0.65	0.65	0.67		0.67
CRUDE FLOW (BPD)	66,321	67,041	5000	3850	56,641	63,083	10,400	0.00	53,305		0.00
CRUDE FLOW (GPM)	1934	1955	150	120	1652	1840	303	0.00	1555		0.00
CRUDE GRAVITY (°API)	46	43	38	41	43	40	43	–	40		–
WATER FLOW (BPD)	75,082	76,205	1,960	2006	64,383	76,538	11,822	0.00	64,664		0.00
WATER FLOW (GPM)	2190	2223	57	58	1878	2232	345	0.00	1886		0.00
WATER S.G.	1.00	0.98	1.02	0.99	0.98	0.98	0.98	–	0.98		–

STREAM NO.	12	13	14	15	16	17	18	19	20	21	22
DESCRIPTION – 60,000 BPD CASE											
TEMPERATURE (°F)	150	150	150	150	85	150	150	150	150	150	75
PRESSURE (PSIG)	200	1900	625	625	1900	200	200	200	200	0.00	200
GAS FLOW (MMSCFD)	5.2	26	2.5	2.5	167.1	5.3	2.6	0.00	0.00	0.00	0.1
GAS S.G.	0.77	0.65	0.67	0.67	0.65	0.78	0.78	0.78	0.78	1.25	0.72
CRUDE FLOW (BPD)	62,136	0.00	9766	9766	231	62,520	0.00	31,215	43	4	105
CRUDE FLOW (GPM)	1812	0.00	285	285	7	1824	0.00	910	2	0.00	3
CRUDE GRAVITY (°API)	41	–	40	40	97	41	–	41	41	41	66
WATER FLOW (BPD)	76,630	0.00	11,873	11,873	53	76,783	0.00	1663	36,733	36,717	56
WATER FLOW (GPM)	2235	0.00	346	346	2	2240	0.00	48	1072	1071	2
WATER S.G.	0.98	–	0.98	0.98	1.01	0.98	–	0.98	0.98	0.98	1.01

Figure 13-6. Process flowsheet.

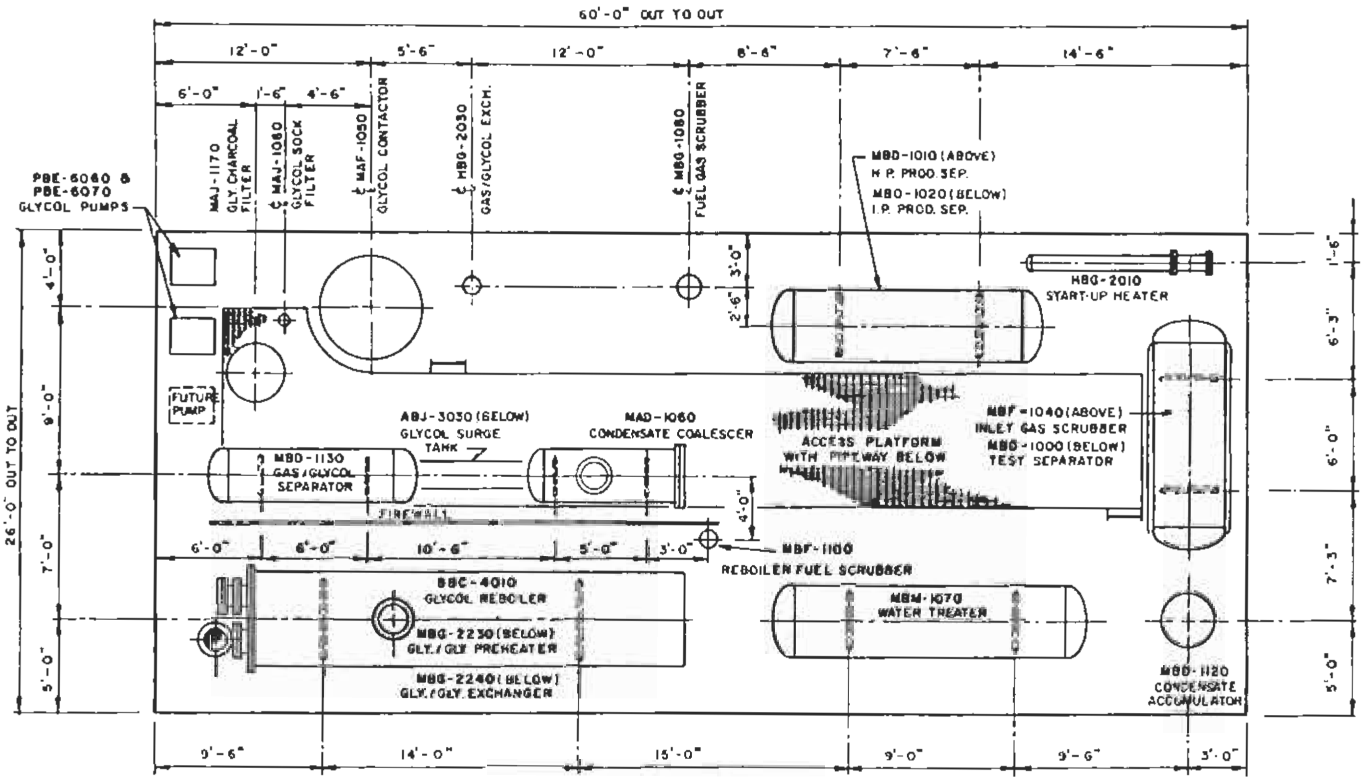


Figure 13-7. Production skid layout.

information is received that affects it. Revisions may be necessary due to a change in scope or a realization that the amount of work was over- or underestimated.

One of the goals of this estimate is to enable the tracking of project cost relative to targets set by the AFE. For this reason, the preliminary cost estimate must be made in a listing that considers the plan of execution. An example is shown in Figure 13-8. As each bid item is awarded, its cost can be compared to the AFE amount for that item to determine if the project is on budget. In addition, this type of cost accounting enables the building of a data base for future jobs that share work items similar to the job in question.

Plan of Execution

A plan of execution for a project begins when the first information is received. This plan must consider the alternatives for breaking the job down into individual work items to be bid out. It must balance time and ease of management against cost for such decisions as the scope of work to include in individual work items; the degree of engineering to perform prior to bid; potential suppliers' work load, capability, and competitive situations; and operator's sole source preferences. Since these are different at different times and for different overall project objectives of size, cost, and timing, each project will have its own optimum plan of execution.

A major problem to be solved is that of scheduling the many activities involved so that the combined efforts of manpower and facilities are directed toward a common goal with coordinated timing. A typical plan of execution is shown in Figure 13-9. The schedule assures coordination of the activities and portrays the overall timing requirements of each work item of the project. The schedule is updated periodically to show the progress that each key item has made relative to "target" dates. This is then used as a tool for directing effort to those areas that are critical to project startup.

For large projects it may be necessary to keep track of the schedule using one of the many standard project scheduling computer techniques. Care must be exercised not to provide so much detail to the computer that critical items get lost in the mass of output. The engineer responsible for each work item must have sufficient detail to be able to track his target dates. The project engineer only requires sufficient detail to assure that these targets remain coordinated.

Basis of Estimate:
200' Water Depth
3 Wells—6 Completions—Unmanned
30 MMSCFD W/1 BBL Condensate/MMSCF
No Dehydration Required—Future Compression
Condensate to be injected into gas p/l
6500' 8'' Pipeline to hot tap in block
Summer 1990 Installation (Actual installation of platform in summer and pipeline in winter)

	Original AFE	Final Cost 1990
1. Equipment		
a. Manifold	$ 50,000	$ 41,280
b. Production Skids	240,000	267,582
c. Gas Sales Meter Skid	60,000	114,586
d. CPI Unit	10,000	10,350
e. Office/Warehouse	15,000	16,335
f. Crane	70,000	77,077
g. Generator	15,000	12,177
h. Freight and Miscellaneous	45,000	35,613
Subtotal	$ 505,000	$ 575,000
2. Instrumentation		
a. Control Panels	$45,000	$ 33,700
b. Installation	55,000	56,300
Subtotal	$ 100,000	$ 90,000
3. Fabrication & Interconnect		
a. Jacket and Anodes	$ 540,000	$ 554,285
b. Piling	330,000	376,222
c. Deck	340,000	405,884
d. Interconnect—Piping & Electrical	235,000	249,081
e. NDT	25,000	11,911
Subtotal	$1,470,000	$1,597,383
4. Offshore Installation		
a. Platform Installation	$ 600,000	$ 510,000
b. Hookup	110,000	150,000
Subtotal	$ 710,000	$ 660,000
5. Pipeline (final length = 14,500 ft)		
a. Materials	$ 150,000	$ 235,000
b. Installation	745,000	1,180,000
Subtotal	$ 895,000	$1,415,000
6. Engineering, Construction Management & Inspection		
a. Engineering & Drafting	$ 220,000	$ 235,000
b. Project Management & Inspection	165,000	130,900
c. Soil Report	35,000	34,100
Subtotal	$ 420,000	$ 400,000
Project Subtotals	$4,100,000	$4,737,383
Contingency	400,000	162,617
Project Totals	$4,500,000	$4,900,000

Figure 13-8. Example project cost summary.

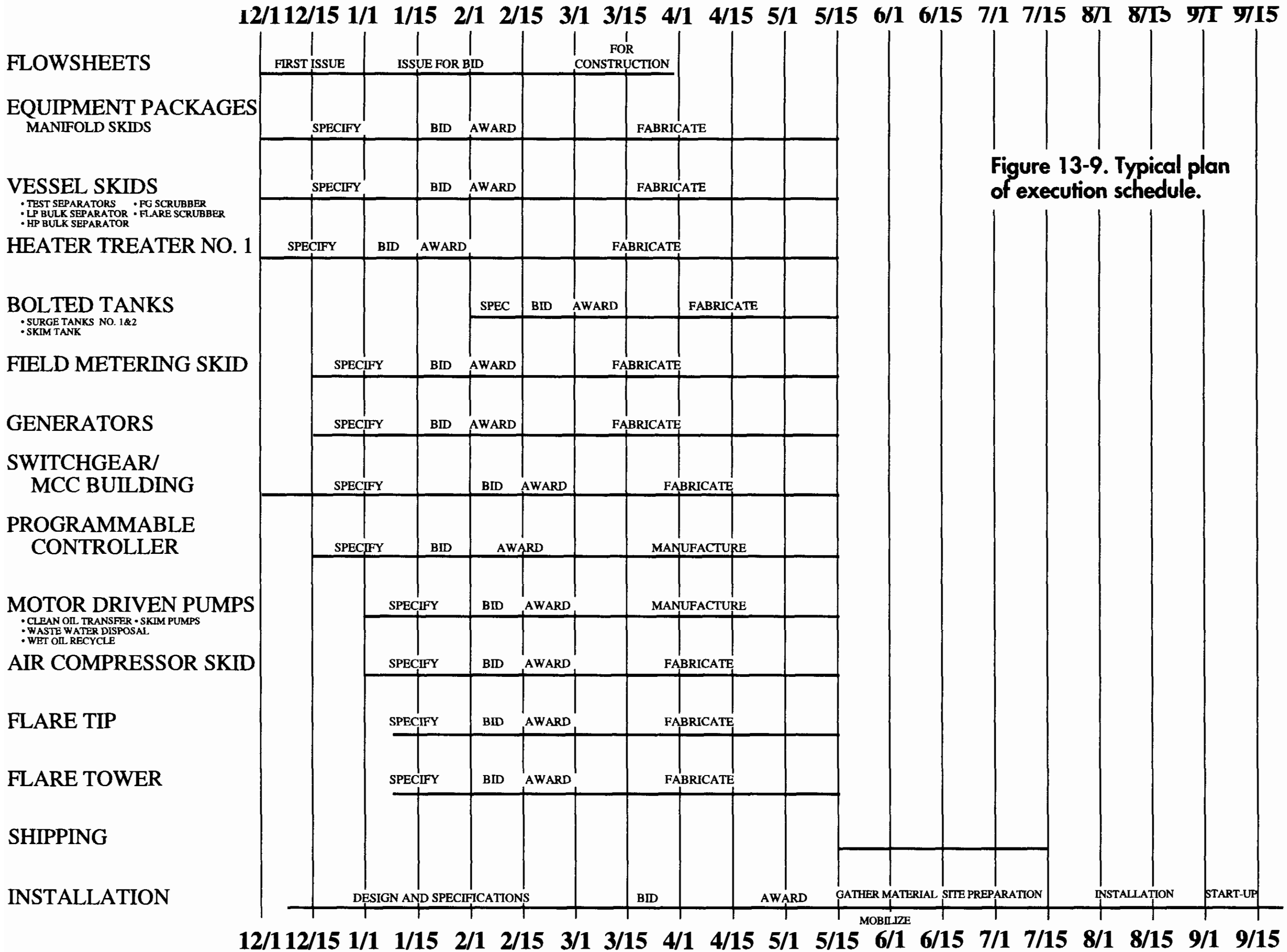


Figure 13-9. Typical plan of execution schedule.

Design Engineering

Once the need and extent of engineering assistance required is determined, design engineering must begin so as to translate the process flowsheets into specific objectives, and determine the activities that will be required to attain these objectives. The basic item upon which all other activities depend and which must be completed early in an engineering design project, are the mechanical flowsheets. In addition, specifications for long delivery items must be written at this stage.

Mechanical Flowsheets

The mechanical flowsheets are established from the process flowsheets. They show every piece of equipment for the entire facility including the process, utilities, firewater system, safety systems, spare equipment, etc. (see Figure 13-10). Every instrument, valve, and specialty item is shown schematically. All items are individually identified with a tag number. Piping should be shown with flow arrows and line numbers indicating size, pressure rating, service, heat tracing, and insulation. In this manner it is possible to reference individual specifications, purchase orders, inspection reports, invoices, etc., to the correct piece of equipment as the job progresses.

Before the mechanical flowsheets can be finalized, a pipe, valve, and fitting specification must be agreed upon. This would specify, for various pressure ratings and service, the pipe wall thicknesses, end connections, branch connections, minimum sizes, instrument and drain connections, etc.

Vessel and Equipment Specifications

The vessel and equipment specifications are established for long delivery items to expedite both the design and the purchasing effort. Every facility is designed for a specific function. The criteria by which the equipment is designed are applicable to the conditions of that facility only. Therefore, some equipment are not typical, off-the-shelf items. However, an experienced engineer will maximize the use of standard items to minimize cost and delivery time.

For vessels important information such as size, nozzle number and size, and required internals must be specified. Figure 13-11 illustrates a typical vessel data sheet.

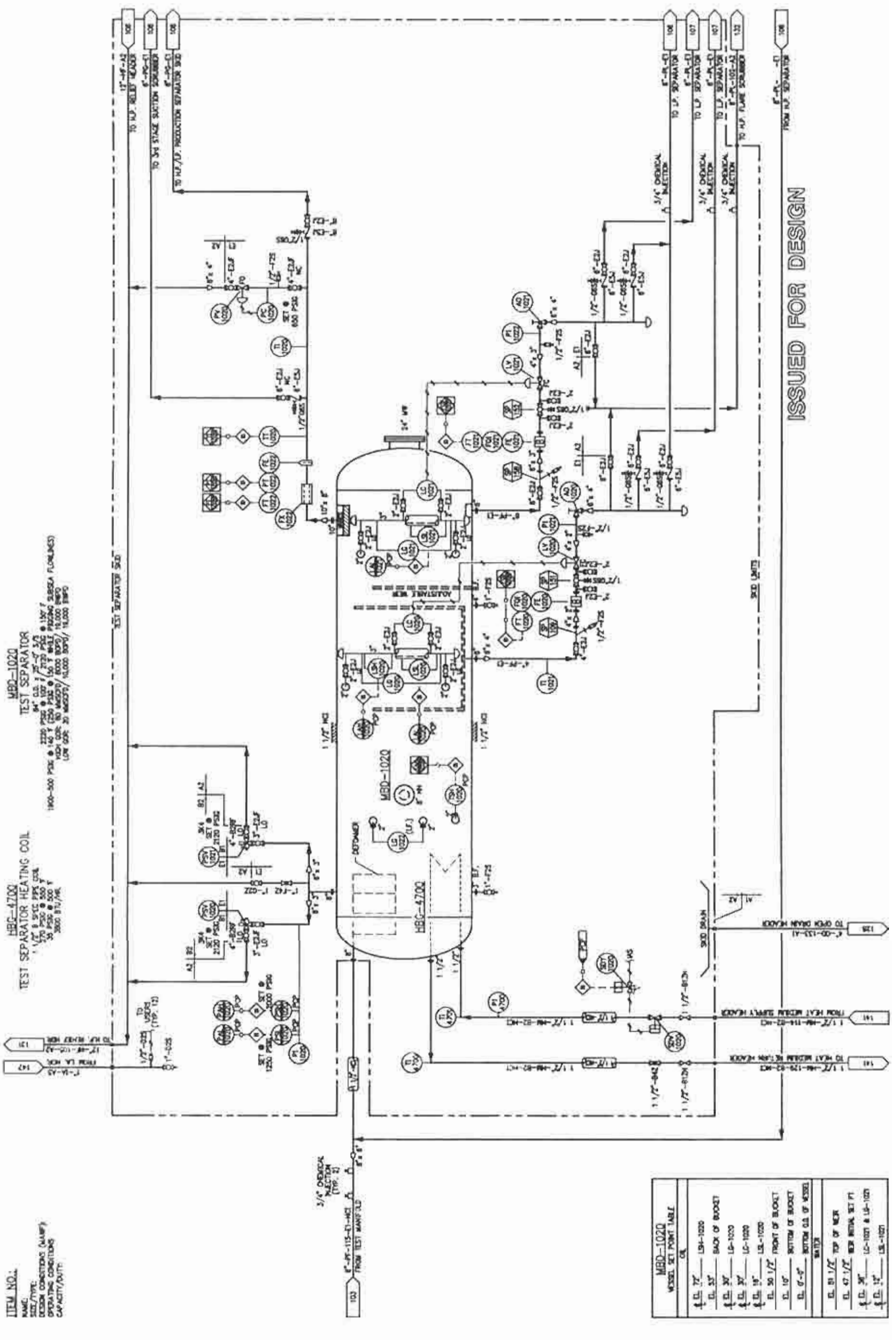
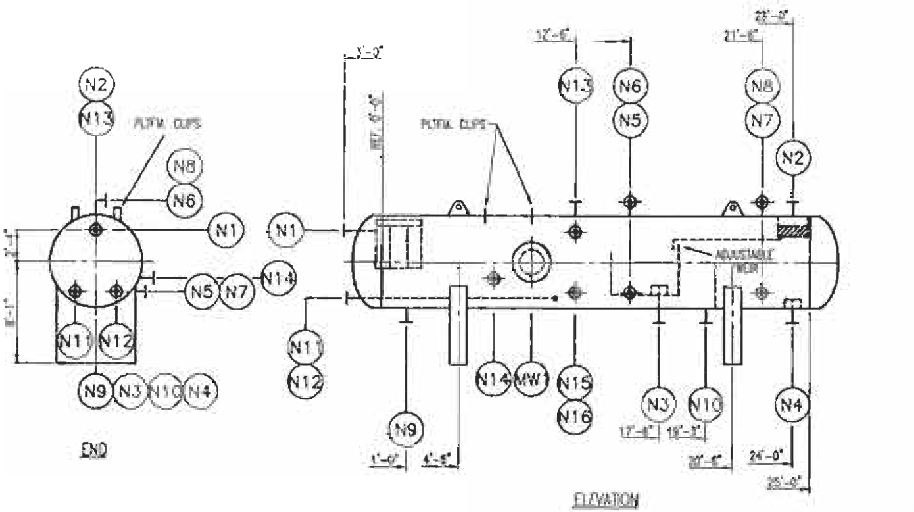


Figure 13-10. Mechanical flowsheet.



ITEM TEST SEPARATOR ITEM No. MBD-1000
No. REQ'D. ONE JOB No. 96124
PURCHASE ORDER No. DATE:

DESIGN AND FABRICATION DATA

Construction To Be in Accordance With the latest Edition Of The ASME Code & Addenda SECTION VIII DIVISION 2
Code Symbol: Required
Design Pressure 2200-2120 PSIG at 100-150 °F
Operating Pressure 900-1800 PSIG at 120-140 °F
Stress Relieve: Yes / No Radiograph: 100%
Joint Eff. - Shell 100 % Heads 100 %
Corrosion Allowance - Shell 0.125" Heads 0.125"
Material: Shell Heads
Flanges Pipe
Studs Nuts
Gaskets
Saddles YES Lugs YES
Hinges Davits Required For Manholes YES
Insulation Thickness 1 1/2" CALCIUM SILICATE Insulation Rings YES - AR
Ladder Clips YES Platform Clips YES
Paint NONE
Demister YES - VANE
Notes: 1) VESSEL TO BE EQUIPPED WITH INTERNAL HEATING COIL 1 1/2" O.D. x 0.281" W.T., A-106 Gr. B; MAWP = 575 PSIG @ 300°F
2) VESSEL TO BE EQUIPPED WITH PORTATEST FOAMFREE SEPARATOR ON INLET NOZZLE
3) ALL VESSEL CONNECTIONS TO BE LWN TYPE FLANGES.

MK	No.	Size	Ser. & Fac.	Type	Service	Proj.
MW	1	24"	900#	RTJ	MANWAY w/ BLD. FLG & DAVIT	
N	16	2"	900#	RTJ	INTERFACE BRIDLE - BOTTOM	-
N	15	2"	900#	RTJ	INTERFACE BRIDLE - TOP	-
N	14	2"	900#	RTJ	TSH	-
N	13	4"	900#	RTJ	PSV	
N	12	1 1/2"	300#	RF	HEAT MEDIA - OUTLET	-

CONNECTION SCHEDULE

MK	No.	Size	Ser. & Fac.	Type	Service	Proj.
N	11	1 1/2"	300#	RF	HEAT MEDIA - INLET	-
N	10	4"	900#	RTJ	DRAIN	
N	9	3"	900#	RTJ	DRAIN	
N	8	2"	900#	RTJ	WATER LEVEL BRIDLE - TOP	-
N	7	2"	900#	RTJ	WATER LEVEL BRIDLE - BOTTOM	-
N	6	2"	900#	RTJ	OIL LEVEL BRIDLE - TOP	-
N	5	2"	900#	RTJ	OIL LEVEL BRIDLE - BOTTOM	-
N	4	6"	900#	RTJ	WATER OUTLET	
N	3	6"	900#	RTJ	OIL OUTLET	
N	2	10"	900#	RTJ	GAS OUTLET	
N	1	10"	900#	RTJ	INLET	-

CONNECTION SCHEDULE

Figure 13-11. Vessel data sheet.

Detailed Engineering

Once the design engineering is completed and the major items of equipment have been specified and bid out, the next step in the project is to perform the detailed engineering. This consists of piping drawings, structural drawings, electrical one-line drawings, instrument data sheets, and control schematics.

Pipe Drawings

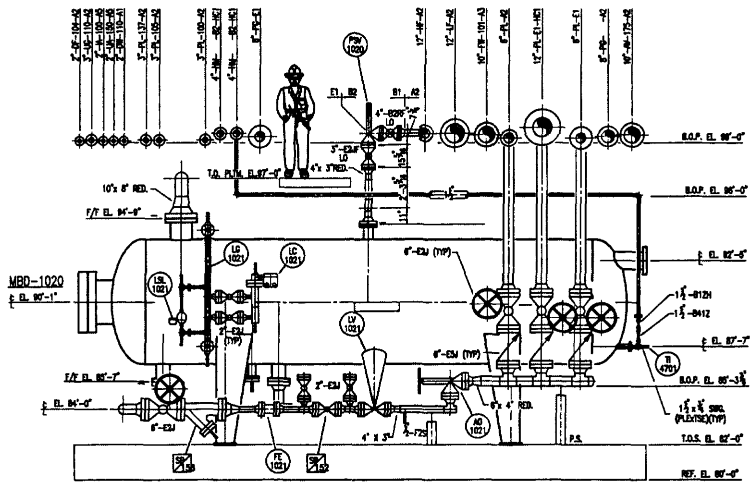
The piping drawings translate to the fabrication contractor the piping arrangement as defined in the mechanical flowsheets. These drawings are usually very simple for onshore facilities, whereas for complex offshore facilities, where space is important, they become very complex. In many cases a good set of piping drawings is the key to a facility that is easy to build and operate. In all cases, a good set of piping drawings is required to speed installation and keep the cost for "extras" to a minimum. Figure 13-12 is an example of a piping drawing.

Structural Drawings

Structural drawings for an onshore facility detail the foundation site development and road work required, as well as any pipe supports or skids for the production equipment. Structural drawings for an offshore facility can include platform drawings, as well as those for production skids themselves. The skids could be installed on wooden or concrete piles, steel or concrete barges, or steel jackets with steel decks. Figure 13-13 is a typical structural drawing for an offshore facility.

The barge, platform, and pile drawings are the first drawings to be made. These drawings are based on environmental wind, wave, and current forces. The structural engineer must be furnished deck space requirements and dead and live loadings. Those loads created by the drilling operations, as well as the loads required for the facilities, must be considered. When all factors that influence design are established, the drawings can be carried out.

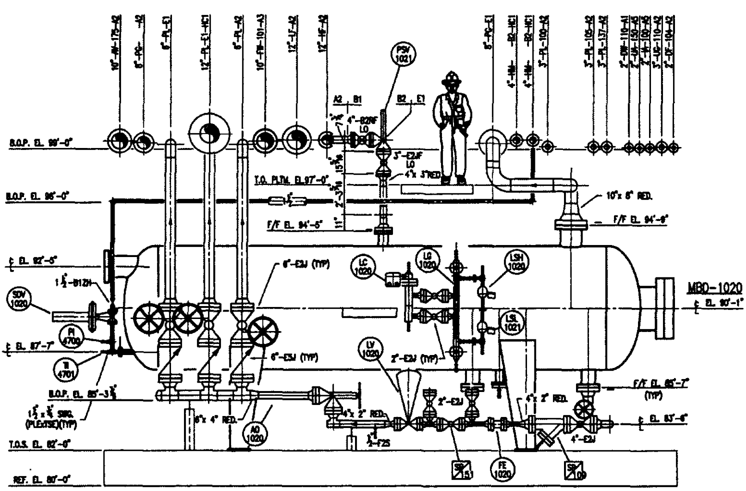
Many times the drilling equipment will be removed before the production equipment is set. The drilling equipment is packaged and skid mounted and the principal components of the superstructure will be main load bearing members boxed together on a spacing compatible with the skidded rig components. Production equipment usually does not involve



SECTION A 528/529

NOTES:

1. DIMENSIONS AND CALLOUTS ARE TYPICAL FOR BOTH THE EAST & WEST SIDES, UNLESS NOTED.
2. PS = PIPE SUPPORT



SECTION B 528/529

REFERENCE [redacted]:
98124-01-1[redacted] P & I D
98124-01-1[redacted] P & I D
98124-03-528 PIPING PLAN - LOWER LEVEL
98124-03-529 PIPING PLAN - UPPER LEVEL

Figure 13-12. Piping drawing.

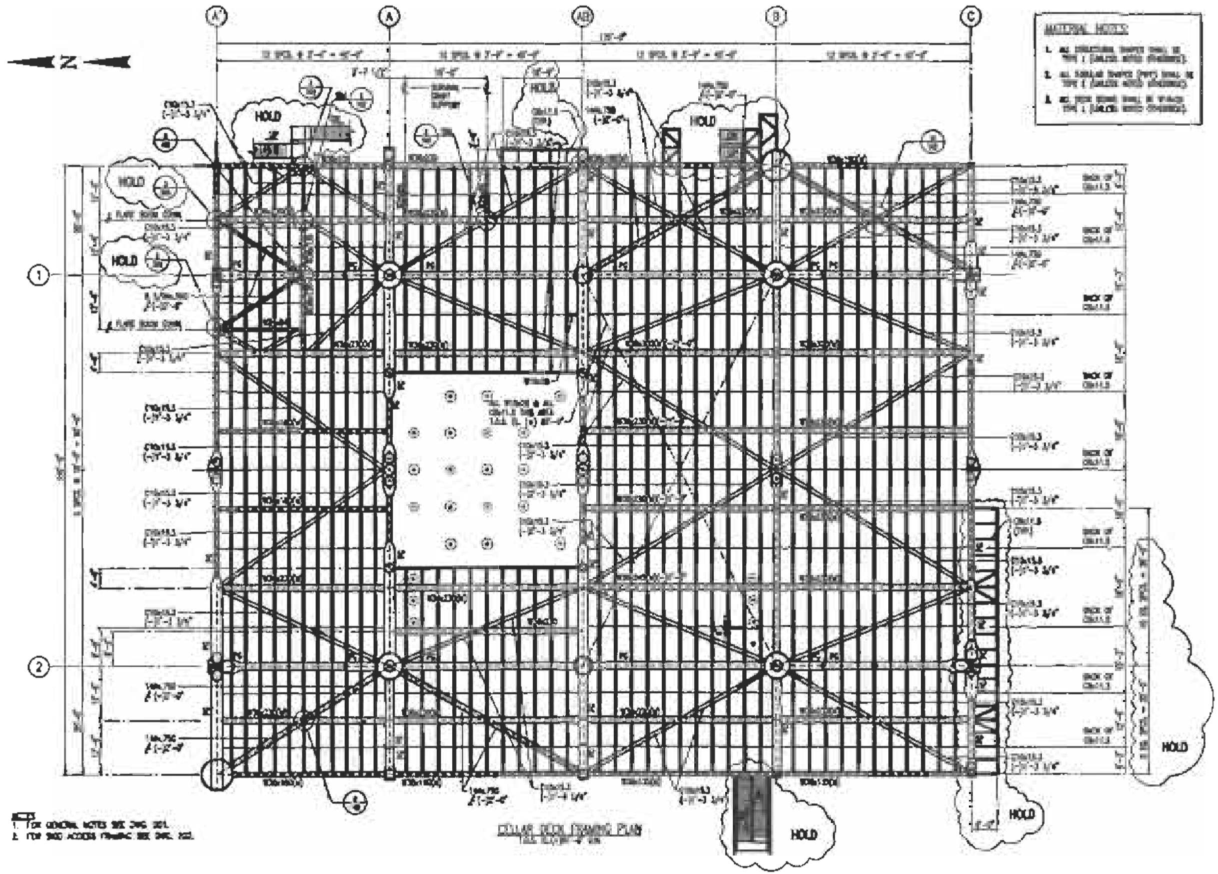


Figure 13-13. Structural drawing.

greater weights than those imposed by drilling components. An exception may occur if large storage is required or very large process vessels are stacked one above the other. More total space may ultimately be required for production equipment than for drilling equipment, but this is designed into the base structure and simply is not occupied during drilling operations.

The superstructure design documents cannot be completed until the size, weight, and location of the production equipment are fairly well defined. If production equipment is to be installed after the wells are drilled, it may be necessary to set a higher value than would be anticipated with completion of the design engineering phase.

Electrical One-Line Drawings

Electrical one-line drawings show the main flow of power to major pieces of equipment. Items shown should include transformers with ratings, circuit breakers with ratings, generators with ratings, motors with horsepower and ratings, relays, buses with voltage rating shown, control stations, and names of equipment as shown on mechanical flowsheets. Figure 13-14 is a typical electrical one-line drawing.

Instrument Data Sheets

Instrument data sheets and control schematics vary considerably in format, depending on company preference. Instrument data sheets usually follow the Instrument Society of America (ISA) format. As shown in Figure 13-15, the instrument data sheets list all necessary information required to specify and identify each instrument. This information may include such items as tag number, operating conditions, materials of construction, manufacturer, and model number.

Procurement

A large part of the "engineering" effort is involved in bidding, evaluating, expediting, and coordinating vendors and vendor information. This is true whether the work is performed by the operator, his engineering consultant, or a "turnkey contractor."

Special equipment, vessels, and instrumentation are usually ordered as soon as the specification can be written, checked, bids obtained and evaluated, and the purchase order issued. Every effort should be made to

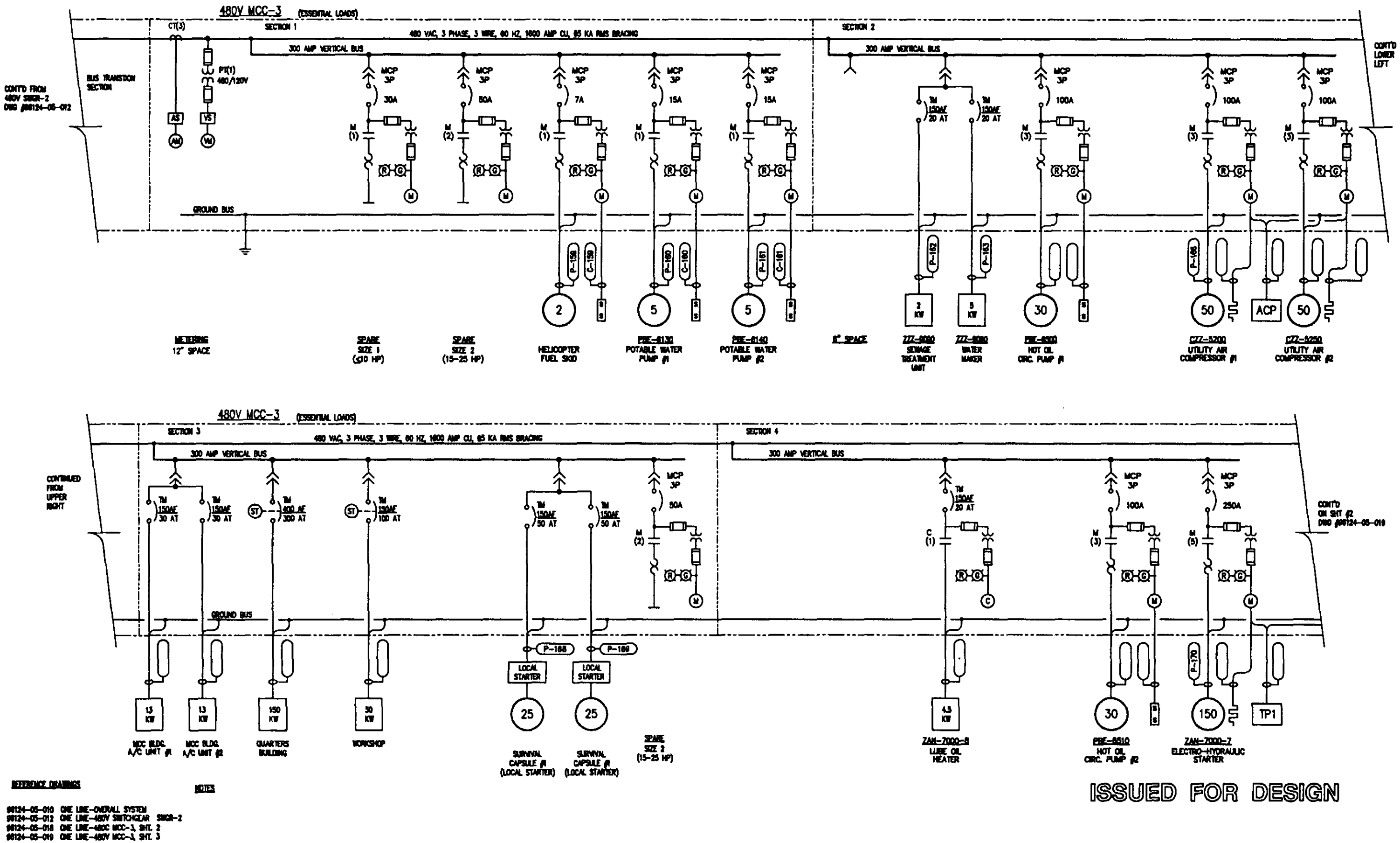


Figure 13-14. Electrical one-line drawing.

PARAGON ENGINEERING SERVICES, INC.	CONTROL VALVES	SHEET 1 OF 7
	NO. BY DATE REVISION	SPEC. NO. 1-CV-01 / REV. D
	A GMH 12/96 ISSUE FOR REVIEW	CONTRACT 96383 / DATE 01/29/96
	B GMH 1/97 ISSUE FOR QUOTE	REQ. 105 / P.O. 960630105
	C GMH 2/97 ISSUE FOR PURCH	BY GMH / CHK'D MEW / APPR.
	D GMH 2/97 GENERAL REV	

GENERAL	1	Tag No.		LV-1011		FV-1011	
	2	Service		SLUG CATCHER INLET (LIQUID)		SLUG CATCHER INLET (GAS)	
	3	Line No./Vessel No.		MBF-1000		MBF-1000	
	4	Line Size/Sched. No.		24"-PG-103-E2 / SCH 80		18"-PG-103-E2 / SCH 80	
	5	Function		SLUG CATCHER INLET		SLUG CATCHER INLET / LEVEL CRL.	
BODY	6	Type of Body		GLOBE		GLOBE	
	7	Body Size	Port Size	12"	10"	12"	10"
	8	Guiding	No. of Ports	CAGE	ONE	CAGE	ONE
	9	End Conn. & Rating		12" - 900 RTJ		12" - 900 RTJ	
	10	Body Material		WCB STEEL		WCB STEEL	
	11	Packing Material		SINGLE TFE		SINGLE TFE	
	12	Lubricator	Iso. Valve	NONE	NONE	NONE	NONE
	13	Seal Type		V-RING		V-RING	
	14	Trim Form		LINEAR (TRIM # 77)		LINEAR (TRIM # 77)	
	15	Trim Material		316 SS W/ CoCr HARDENED TRIM		316 SS W/ CoCr HARDENED TRIM	
	16	Seat Material		316 SS W/ CoCr HARDENED TRIM		316 SS W/ CoCr HARDENED TRIM	
	17	Required Seat Tightness		ANSI CLASS III		ANSI CLASS III	
	18	Max Allow Sound Level dBA		85		85	
ACTUATOR	19	Type of Actuator		PNEUMATIC DIAPHRAGM		PNEUMATIC DIAPHRAGM	
	20	Valve Fail Position		OPEN		CLOSED	
	21	Supply to Actuator Normal / Max.		6 - 30 PSIG		6 - 30 PSIG	
	22	Self Cont.	Ext. Conn.	N/A	N/A	N/A	N/A
	23	Diaphragm Material		MFG. STD.		MFG. STD.	
	24	Diaphragm Rating		55 PSIG		55 PSIG	
	25	Spring Range		BENCH SET 8-24 PSIG		BENCH SET 11-30 PSIG	
	26	Manual Overide		N/A		N/A	
	27	Actuator Mfg. / Model No.		FISHER 657-4 SIZE 87-4 (1R6760)		FISHER 667-4 SIZE 87-4 (1R6760)	
ACCESSORIES	28	Filt. Reg.	Supply Gage	67 FR	YES	N/A	N/A
	29	Positioner		3582I (4-20 MADC IN, 6-30 PSIG OUT)		N/A	
	30	Pilot Valve		TRAVEL STOP (SEE NOTE 4)		TRAVEL STOP (SEE NOTE 4)	
	31	Limit Switches		N/A		N/A	
	32	Solenoid Valve		N/A		N/A	
	33	Position Transmitter		N/A		N/A	
SERVICE	34	FLOW UNITS					
	35	Fluid		NAT. GAS / CONDENSATE (NOTE 3)		NATURAL GAS	
	36	Quant. Max.	Cv	(SEE NOTE 1)	37985 / 1754	400 MMSCFD	37985
	37	Quant. Oper.	Cv	(SEE NOTE 1)	37985 / 1754	400 MMSCFD	37985
	38	Valve Cv	Valve FL	62500 / 1970	K = 0.64	62500	C1 = 31.7
	39	Norm. Inlet Press.	D P	1250 PSIG	15 PSID	1250 PSIG	15 PSID
	40	Max. Inlet Press.		1800 PSIG		1800 PSIG	
	41	Max. Shut Off	D P	1800 PSI	1800 PSI	1800 PSI	1800 PSI
	42	Temp. Max.	Operating	100 DEF F	70 DEG F	100 DEF F	70 DEG F
	43	Oper. sp. gr.	Mol. Wt.	0.65 / O.56	18.7 / N/A	0.65	18.7
	44	Oper. Visc.	% Flash	0.014 / 0.1734	3.8%	0.014	N/A
	45	% Superheat	% Solids	N/A	N/A	N/A	N/A
	46	Vapor Press.	Crit. Press.	1200 PSIA	1600 PSIA	1200 PSIA	1600 PSIA
	47	P&ID No.		SD-43244		SD-43244	
	48	Manufacturer		FISHER		FISHER	
	49	Model No.		12" - 657-EHD		12" - 667-EHD	

Notes: 1.) FLOW REQUIRED 400 MMSCFD AND 8300 GPM CONDENSATE.
2.) "*" DENOTES INFORMATION SUPPLIED BY VENDOR.
3.) NORMALLY, FV-1011 & LV-1011 ARE IN THE OPEN POSITION. DURING SLUGGING CONDITION FV-1011 WILL CLOSE AND LV-1011 WILL CONTROL LIQ. LEVEL.
4.) VALVE TO BE SUPPLIED C/W MINIMUM CLOSE TRAVEL STOPS.
5.) FV-1011 & LV-1011 WILL EXPERIENCE THE IMPACT OF LIQUID SLUGS (SG=0.56) ARRIVING AT VELOCITIES UP TO 27 FPS WHILE IN ANY POSITION FROM FULLY OPEN TO FULLY CLOSED.
6.) TAG ITEMS TO BE SUPPLIED WITH SS TAGS PERMANENTLY ATTACHED.

ISA Form S20.51A

Figure 13-15. Instrument data sheet.

ensure that sufficient detail is incorporated into the specification. The specification will usually include references to engineering standards that have been developed to reduce the amount of repetitive writing and to provide uniformity.

In writing specifications, the following items must be considered and statements included if applicable:

1. Scope of the bid
2. Contractor's responsibility and scope of work
3. Design conditions
4. Company furnished materials
5. Bid details
6. Timing requirements
7. Inspection and testing
8. Acceptance
9. Preparation for shipment
10. Warranty
11. Change orders and extra work
12. Compensation schedules
13. Contractor furnished data

In addition, engineering specifications, data sheets, and drawings must be furnished listing applicable codes, materials, techniques, etc. Figure 13-16 shows some typical engineering standards that could be included in a specification.

Bid Evaluation

Evaluation of bids is the process of breaking down the vendor's quotation into its various components, items, or features and comparing them with the original specifications. This breakdown is recorded on a tabular form. The most difficult and time-consuming part of making a tabulation is usually the listing of the subjects for comparison. This step may be simplified if the specific requirements stated in the inquiry are listed on the tabulation form as the standard for comparison. The information from the quotation is then posted adjacent to these headings so that omissions and differences between the quotes and the specification are easily observed. Evaluation of quotations normally should not begin until all of the anticipated quotes are received.

All bids should be opened at one time to minimize any possibility of one bidder receiving an unfair advantage. While it is possible to play one bid-

Structural:
- Offshore Platform Fabrication
- Structural Steel Skid-Mounted Assemblies
- Offshore Platform Installation
- Offshore Steel Fabrication
- Concrete Barge and Platform

Equipment:
- Unfired Pressure Vessels
- Reciprocating Pumps
- Diesel Cranes
- Pneumatic Cranes
- Firewater Pumps
- Air Compressor Packages

Piping:
- Piping, Valves, and Fittings
- Valve Tables for Piping, Valves, and Fittings
- Material for Piping, Valves, and Fittings
- Piping/Instrument Specialty Items

Electrical:
- General Electrical Installation and Testing
- Generator Sets
- Electrical Heat Tracing
- Motors

Instrumentation:
- Instruments and Instrument Installation
- Facility Master Control Panels
- Combustible Gas Detection Systems

Pipelines:
- Offshore Pipeline Installation
- Marsh Pipeline Installation
- Pipe Coating: Thin Film, Epoxy
- Pipe Coating: Coal Tar Enamel (Yard Applied)
- External Coating and Vulcanized Coating
- Offshore Pipeline Construction Survey
- Underwater Diving Inspection
- Offshore Pipeline Anodes
- Underwater Check and Ball Valves
- Launcher/Receiver Fabrication

Protective Coatings:
- General
- Insulation

Figure 13-16. Example list of engineering standards.

der against the other ("bid shopping"), any experienced project engineer knows that in the long run this will be counterproductive. If vendors expect the work to be given to the low bidder they will put their best efforts into the prices. If they expect an "auction," their bid prices will reflect this.

Purchase Order

The purchase order is issued to the successful bidder after all quotations have been evaluated. It includes all information of the inquiry and, in addition, specifies conditions of purchase, delivery, consignment instructions, etc.

Inspection and Expediting

An important phase of the project is inspection during the fabrication and construction activities. It is the inspector's responsibility to assure that the finished equipment or material is of acceptable quality and complies with all requirements of the purchase order. The inspector witnesses tests on mechanical equipment such as pumps and compressors, observes and approves fabrication methods of vessels, pipe, and structural steel, and generally ensures that the specified workmanship is being performed on the purchased equipment.

On large projects, it may be necessary to provide an expediter to assure delivery of purchased materials and equipment at the job site. Bulk material such as valves and fittings are normally not a problem as they are usually available from a supplier's warehouse. However, the progress of specially designed, long delivery, or otherwise critical equipment must be followed continuously by expediters. Manufacturers will estimate probable delivery dates for equipment to be fabricated in their shops, but these estimates depend on prompt delivery of material from their suppliers, efficient scheduling of shop work and early receipt of drawings from the purchaser. The purchaser's expediter can do much to assure that estimated delivery dates will be met by working with both the manufacturer and his own organization. The expediter should seek and study all information that might affect delivery, anticipate delays or bottlenecks, and resolve these with the vendor. He should assist the vendor in solving his procurement problems with sub-suppliers. If delivery schedules must change he should advise his employer as early as possible. On small projects these functions are performed by the inspectors.

Startup

Although the actual startup is properly the responsibility of the operating and not the engineering staff, there is considerable engineering input and planning that should go into the process. Planning is necessary to assure that all systems are checked out to the fullest extent possible, that the right type and sufficient quantities of expendable material (e.g., glycol, lube oils, greases, heat medium) are on hand, replacement parts (especially gaskets, sealing material, and filters) are available if needed, etc. Finally, a procedure must be developed to "purge" the process. That is, to take it from its condition of being full of air through the explosive limit until it becomes full of natural gas and other hydrocarbons.

It is best if the operating personnel responsible for the startup participate in this planning process with the help of engineering. The conscientious preparation of a startup plan is one of the best ways to learn the facility. Often, if the startup plan is done early enough, areas where minor design changes will increase operability will be found.

While startup is the responsibility of operations, the design engineering staff should assist in an advisory role. Often, the designers are best able to recommend alternatives if a problem develops or the startup plan must be modified on the spot. In addition, the experience gained from startup will help the engineering staff in designing safer, more practical facilities in the future.

A startup manual may contain only a few pages or may take the form of a book. In a complex facility it may describe the following:

1. Overall purpose and design of the installation
2. Operation of the process
3. Details and operating descriptions of
   (a) Overall instrumentation system (pneumatic and/or electronic)
   (b) Electrical systems
   (c) Data transmission system
   (d) Utility and firewater systems
   (e) Any other system
4. Instructions for installation of equipment
5. Purge and preparation for operation
6. Procedure for startup

Dossier

The startup manual is sometimes erroneously confused with the dossier. Their functions are generally distinct, although their coverages overlap slightly. The dossier is a compilation of reduced size drawings of the project, and all manufacturers' bulletins, manuals, code reports, material certifications, etc., for all purchased equipment in the project. The most common inclusions are reduced-sized drawings of flowsheets, vessel internals, control system schematics, and the SAFE chart.

The startup manual often refers to the dossier for specific instructions, rather than duplicating them. The dossier rarely refers to the startup manual.

Operating instructions of purchased items should be in the dossier, not the startup manual. Critical items from purchased item bulletins should be relisted in the startup manual in outline form for emphasis.

PROJECT CONTROL AND EXECUTION FORMAT

The formal start of a project is considered to occur at the "kick-off" meeting where, for the first time, the complete project team is brought together and the unified work effort begins. All objectives, plans, schedules, and procedures developed are presented at the meeting and formal work assignments are made. At this point, the organization of the project is essentially complete. Prior to the "kick-off" meeting, most of the project work has centered around the project engineer with certain key people supplying staff and administrative assistance. The "kick-off" meeting marks the point where responsibilities are delegated and the work spreads out in several directions. The role of the project engineer shifts from that of planner and organizer to that of coordinator and controller.

Project Control

Engineering Control

The most important part of project control begins at the outset of the project with controlling the engineering effort. Priorities set and decisions made at this point will affect project timing and cost throughout the job.

Tasks Schedules and Drawing Lists. The first step in setting engineering priorities and determining the numbers of engineers and draftsmen

needed to meet timing objectives is to develop a task schedule for each of the engineering tasks shown on the Plan of Execution. An example schedule of tasks is shown in Figure 13-17. The task schedule shows a week by week or month by month accounting of the engineering and drafting effort by discipline required to meet the targets in the plan of execution.

To help in estimating manhours, it is necessary to develop a drawing list concurrently with the task schedule. Figure 13-18 shows a typical drawing list. Each drawing necessary to complete the engineering effort is listed, and the drafting manhours necessary to complete the drawing are estimated. The drawing lists and task schedules define the engineering effort from the outset and provide excellent tools with which to measure actual progress.

Manpower Estimate. The task schedules for each identified task can now be added together to determine an overall manpower requirement for the entire project as is shown in Figure 13-19. At this point it may be desirable to adjust the plan of execution to help smooth out manpower loading. A plan of execution that requires large swings in manpower is probably not practical.

A comparison of manhours required with the time available will determine the approximate number of personnel needed. This preliminary estimate will permit evaluation of manpower requirements against availability and will be the basis for appointing key personnel. The levels of manpower required during progression of a project follow a similar trend whether involving an offshore production platform or an onshore processing facility.

Organization Chart. As the manpower requirements become clear, an organization chart should be prepared that clearly shows lines of authority/communication and corresponding responsibilities. There is no universally accepted format for the organization chart, but it should be a "tree" type with logical branching flowing downward from the project engineer.

Figure 13-20 illustrates a typical organization chart for a large engineering project. Obviously, each organization chart must be designed to fit the specific requirements of the project. The types of skills available will also influence the organization chart and how the work is divided. For example, a particular engineer might be well qualified in both instrument and electrical specialties and the nature of the project might be such that these categories of work could be combined into one responsibility.

Equipment List. The equipment list is a tool that aids in coordinating the specifications, procurement, and inspection efforts. As shown in Fig-

Turbocompressor Unit Related Process Equipment
Task Schedule: Engineering Phase

CODE	ENGINEERING	1		2		3		4		5		6		7		8		9		Planned	To Date
E-A	Project Engineer	2	2	25	65	75	66	56	56	56	56	32	15	15	15	6	6	6	6	622	538
E-2	Instrument Engineer	0	0	0	0		10	10	10	14	14	34	34	5	0	0	0	0	0	132	132
E-4	Structural Engineer	0	0				25	35	45	55	45	35	30	10	10	10	10	0	0	310	290
CODE	DRAFTING	1		2		3		4		5		6		7		8		9		Planned	To Date
D-A	Drftng Coordinator					10	10	15	15	15	10	10	9	7	7	7	0	0	0	115	108
D-1	Proc./Mech. Drftng				25	30	45	75	115	125	115	100	80	40	40	35	35	0	0	860	790
D-9	Structural Drftng	0	0	0	0		25	30	45	65	50	35	25	10	5	5	5	5	5	310	290
D-5/7	Elec./Instr. Drftng	0	0	0	0	0	0	25	35	50	20	10	10	5	0	0	0	0	0	155	155
CODE	PROCUREMENT	1		2		3		4		5		6		7		8		9		Planned	To Date
R-1	Buyer	0	2		1	2	15	30	35	45	55	65	70	65	60	55	45	30	30	653	445
R-2	Expediter	0	0	0	0	0	0		10	10	20	20	20	21	25	25	35	25	25	377	126
CODE	PROJECT MGMT	1		2		3		4		5		6		7		8		9		Planned	To Date
P-1	Project Manager	15	25	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	35	840	460
P-2	Cost/Scheduler						20	20	20	20	20	20	20	20	20	20	20	20	20	400	180
O-1	Secretary		2	3	5	10	20	20	30	30	30	30	25	30	30	30	40	40	30	610	265
CODE	CONSTRUCTION MGMT	1		2		3		4		5		6		7		8		9		Planned	To Date
C-4	Construction Coordinator																	30	30	340	0
C-1	Vendor Surveillance											30	30	30	40	60	80	120	120	660	130

COST	DISCIPLINE TOTALS (Month)	1		2		3		4		5		6		7		8		9		Activity	Activity
CODE	(Bi-Weekly)	Aug	Sep	Sep	Oct	Oct	Nov	Nov	Dec	Dec	Dec	Jan	Jan	Feb	Feb	Mar	Mar	Apr	Apr	Planned	To Date
E	Engineering	2	2	25	65	75	101	102	112	125	115	101	79	30	25	16	16	6	6	1,064	960
D	Drafting	0	0	0	25	40	80	145	210	255	195	155	124	62	52	47	40	5	5	1,440	1,343
R	Procurement	0	2	0	1	2	15	30	45	55	75	85	90	86	85	80	80	55	55	1,030	571
P	Project Management	15	27	38	40	45	75	75	85	85	85	85	80	85	85	85	95	95	85	1,850	905
C	Construction Management	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Total MH Planned	17	31	63	131	162	271	352	452	520	470	426	373	263	247	228	231	161	151	5,384	3,779
	MH Actual	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0		

Figure 13-17. Typical task schedule.

Turbocompressor Unit Related Process Equipment
Drawing Progress: Unit–Process Mechanical

Project:
Desc: Client Name

Report: PROG_1
Run: 00/00/97 - 12:00 PM

Drawing No	RV	Description	Pct Comp	Start Plan	Start Act	Issue Plan	Issue Act	Client App Plan	Client App Act	Cert Issue Plan	Cert Issue Act	Comments
			PROCESS/MECHANICAL									
			PROCESS/MECHANICAL									
96341-01-001	B	PROCESS FLOW DIAGRAM	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-100	B	P&ID - LEGEND SHEET	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-101	B	P&ID - 1ST STAGE COMPRESSION	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-102	B	P&ID - 2ND STAGE COMPRESSION	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-103	B	P&ID - DISCHARGE SCRUBBER	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-104	B	P&ID - FUEL GAS	100%	/	/	10/5	10/21	/	/	/	2/7	
96341-01-105	B	P&ID - SEAL OIL RUNDOWN TANKS	100%	/	/	10/5	10/21	/	/	/	2/7	
		Avg Pct Complete, Early Start, Late Finish:	100%									

Figure 13-18. Typical drawing list.

Turbocompressor Unit Related Process Equipment
Design Engineering
Discipline Manpower Schedule

COST CODE	DISCIPLINE TOTALS (Month)	1		2		3		4		5		6		7		8		9	
	(Bi-Weekly)	Aug	Sep	Sep	Oct	Oct	Nov	Nov	Dec	Dec	Dec	Jan	Jan	Feb	Feb	Mar	Mar	Apr	Apr
E	Engineering	2	2	25	65	75	101	102	112	125	115	101	79	30	25	16	16	6	6
D	Drafting				25	40	80	145	210	255	195	155	124	62	52	47	40	5	5
R	Procurement		2		1	2	15	30	45	55	75	85	90	86	85	80	80	55	55
P	Project Management	15	27	38	40	45	75	75	85	85	85	85	80	85	85	85	95	95	85
C	Construction Management											30	30	30	40	60	80	150	150
	Total MH Planned	17	31	63	131	162	271	352	452	520	470	456	403	293	287	288	311	311	301
	MH Actual	18	72	41	145	150	142	201	198	206	89	142	236	333	472	224			
	Accum MH Planned	17	48	111	242	404	675	1,027	1,479	1,999	2,469	2,926	3,329	3,622	3,909	4,197	4,508	4,819	5,120
	Accum MH Actual	18	89	130	274	424	566	767	965	1,171	1,260	1,401	1,637	1,970	2,442	2,665			

	Activity Totals	*Activity To Date*	*Activity Wgt*
Engineering	1,064	960	16.66%
Drafting	1,440	1,343	22.56%
Procurement	1,030	571	16.14%
Project Management	1,850	905	28.98%
Construction Management	1,000	130	15.67%
Total MH Planned	6,384	3,909	100.00%
MH Actual			
Accum MH Planned		4533.50	
Accum MH Actual		2040.08	

Figure 13-19. Overall manpower schedule.

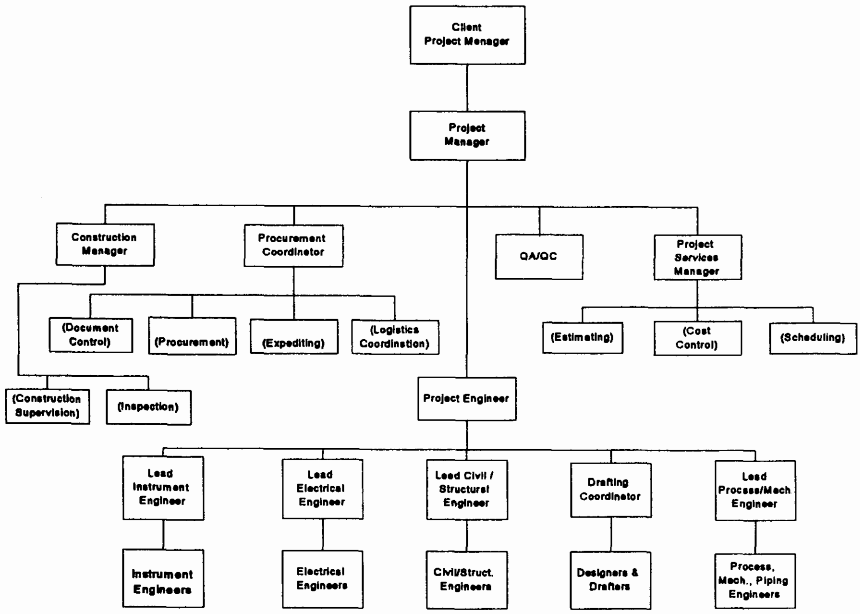


Figure 13-20. Typical organization chart.

ure 13-21, the equipment list is a listing of specific information required for major items. It will list information such as:

1. Process vessels—size, design pressure and temperature.
2. Pumps—type of pump, flow rate, horsepower, differential head, motor bhp.
3. Compressors—type of equipment and driver, driver horsepower, flow capacity, suction and discharge pressures.
4. Generators—type of equipment and driver, driver horsepower, generator output.

Weekly Project Meetings. These meetings can be a valuable aid in helping the project engineer keep his project directed toward its goal. An agenda should be prepared and followed. The project engineer should determine the frequency of the project team meetings and decide which personnel are required to attend. Weekly meetings have proven to be valuable to discuss the following: a brief report of performance to date and hours consumed compared to schedule and estimated hours, a query to members to determine whether their progress is being delayed for lack

Revision: D
Revision Date: 3/21/97
Revision By: MTH

EQUIPMENT LIST

PES Project No. 96124

Tag Number		Description	Size	Capacity	Horse Power	Weight (kips) Dry	Weight (kips) Oper	Remarks
GAY-	0500	Prod.Manif., 16-Slot, 4 Log	29' L x 16'-6" W	200 MMSCFD/60,000 BOPD/30,000 BWPD	N/A	76	76	TCS
MBD-	1000	HP Separator	84" OD x 25' S/S	200 MMSCFD/90,000 BLPD	N/A	128	134	Horizontal, 2 Phase
MBD-	1010	IP Separator	84" OD x 25' S/S	200-30 MMSCFD/90,000 BLPD	N/A	128	134	Horizontal, 2 Phase
MBD-	1020	Test Separator	84" OD x 25' S/S	80 MMSCFD/8000 BOPD/19,000 BWPD	N/A	122	130	Horizontal, 3 Phase Bucket and Weir
MBD-	1030	LP Production Separator	96" OD x 25' S/S	10 MMSCFD/36,000 BOPD/45,000 BWPD	N/A	45	89	Horizontal, 3 Phase Bucket and Weir
MBD-	1040	LP Production Separator	96" OD x 25' S/S	10 MMSCFD/36,000 BOPD/45,000 BWPD	N/A	45	89	Horizontal, 3 Phase Bucket and Weir
ABJ-	2000	Gas Flotation Cell	40' L x 14' W	78,000 BWPD	100	20	80	Wemco (Baker-Hughes)
ABJ-	2010	Slop Oil Tank	15'-6" DIA x 16' H	500 BBL	N/A			1.5 psig MAWP
HBN-	4000	Waste Heat Recovery Unit		20 MMBtu/hr (460 GPM (Tubes))	N/A	155	155	
HBG-	4070	Slop Oil Tank Heater		3,800 Btu/hr	N/A			Install Internal to Tank.
CAE-	5000	Reciprocating Compressor Stage 1		2.6 MMSCFD				Recip.
CAE-	5010	Reciprocating Compressor Stage 2		6 MMSCFD	2000			Recip.
CBA-	5100	VRU Compressor		2.6 MMSCFD	300			Rotary Screw
CZZ-	5200	Air Compressor #1		125 SCFM @ 150 psi	50			Airdyne Rotary Screw
PBE-	6130	Potable Water Pump #1		60 gpm @ 65 psi DP	5			API 610 In–line Centrifugal
PAX-	6300	Oil Sales Pipeline Pump #1		875 gpm @ 1900 psi DP	1500	19	20	BW/IP
PAX-	6310	Oil Sales Pipeline Pump #2		875 gpm @ 1900 psi DP	1500	19	20	BW/IP
ZAN-	7000	Turbine Generator #1	30' L x 8' W	4750 kW		77	77	Solar Taurus 60
ZAN-	7100	Turbine Generator #2	30' L x 8' W	4750 kW		77	77	Solar Taurus 60
ZAN-	7300	Emergency Generator	20' L x 10' W	500 kW		50	54	
ZAU-	9040	LACT Unit	30' x 15'	30,000 BPD/run	N/A	53	58	3 - 50% Runs

Figure 13-21. Equipment list.

of information, and a review of changes of scope and schedule. The weekly meeting permits a sharing of ideas between the team without spending the time for correspondence and numerous individual conversations. To be successful, the meeting must be planned for a definite and specific purpose with only those people attending who can contribute to the discussion. Complete notes of the meeting should be made to record agreements and decisions so that responsibilities are clearly outlined and work assignments completely understood.

Engineering Timing Control. The basis of time control is accurate status reporting. The status report measures performance results, which are then evaluated against the planned schedules. This comparison indicates whether or not corrective action must be taken. If the status report indicates that the project is behind schedule, steps must be taken to advance the work in accordance with the schedule. This might be done in several ways: increase working hours with present team, add manpower, shift work assignments from groups that are ahead of schedule, and/or replace inefficient members. If, on the other hand, the status of the work is ahead of schedule, it may be possible to reduce costs by decreasing overtime or the number of people assigned. This situation is much more difficult to analyze than a lagging job because of the subtle human tendency to fill in work in order to appear occupied, but it is equally important to detect it and take corrective action, to minimize engineering cost.

Engineering Cost Control. Control of engineering is exercised not only by controlling progress but also by the equally important function of cost control. The job status must be accurately determined if time and cost control are to be effective. It is not enough to simply compare manhours expended against the total estimated manhours to gauge the percentage of completion of a project. Actual progress must be measured. For the drafting effort this can be done accurately using the drawing list. By counting the number of drawings completed and estimating the degree of completion for drawings in progress, a comparison can be made against the total drawings that will be required and an accurate drafting status will result.

Engineering, administrative, and other activities with statuses not related to drawing progress can be handled similarly by comparing actual progress and expenditure to that targeted on the task schedule. As a check against the measured figures, all team members should report an independent estimate of their progress and percent completion of their assigned work for comparison. By studying all estimates and reports, the project engineer can be assured of an accurate status picture.

Project Cost Control

The first step to good project cost control is good cost accounting. The main tool for this is the preliminary cost estimate developed during the project definition phase. This document listed budget costs for each work item in the plan of execution. As the project becomes better defined, these costs are periodically updated. Differences can then be explained and total project cost revised even before the first item is purchased. As bids are awarded and commitments are made, the total project amount can be adjusted accordingly.

This method of cost accounting has the additional benefit of helping to provide a data base for future projects. Since actual costs are identified with individual work items (as individual vessels, skid packaging, etc.) they are more easily extrapolated to arrive at estimates for future jobs.

While accounting for costs can be somewhat routine, controlling cost is an entirely different matter. Like the federal budget, once a project is committed a large part of the final cost is beyond the control of the project engineer. It is not unusual for bid items to vary significantly from time to time due to contractor work load, weather conditions, and availability of equipment. In addition, many times the difference in price between the low bidder and the next low bidder could be as much as 10% to 20%. If the low bidder had not been included in the bid list, the item cost would be that much higher.

A large part of cost control occurs during the engineering phases. While controlling engineering costs themselves are important, there are other decisions made in both the design and detail engineering phases that may have a much greater impact on project costs. These can be broken down in the following categories:

Design Philosophy. A recent study* of differences between typical North Sea and Gulf of Mexico designs for a given flow rate indicated that one-third of the greater cost for North Sea designs was due to engineering decisions made early on in the project life. These had nothing to do with size, European construction cost premiums, governmental regulations, or offshore installation premiums.

Engineering Details. The technical specifications for any job contain untold opportunities for costs to get out of control. While any one decision may be minor in cost premium, the overall effect could be surpris-

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*Paragon Engineering Services Inc., Houston, Texas.

ing. One company had the opportunity to bid out the identical line heater/separator production skid with two different pipe, valve, and fittings specifications, both of which satisfied all applicable codes and standards. A cost difference of 12% on the total skid price (including vessels and instrumentation) resulted.

Scope of Work Definition. The scope of work should be clear so that the true low bidder can be identified. Often, the apparent low bidder has taken exceptions to the scope of work or quality of supply that makes another contractor the actual low bidder. This is particularly true with bids involving some degree of engineering. An example would be a compressor package where different engine and/or compressor options must be evaluated.

Extra Work. Of perhaps more importance is the enforcement of contract terms. On almost every major contract, the contractor will test the inspector's interpretation of an appropriate extra work item. A procedure for determining allowable extra work must be included in the award specification and must be policed rigidly by the inspector if costs are to be controlled.

Choice of Acceptable Bidders. There is never a lack of people who are willing to bid on any item. By limiting the bid list to people who are experienced and commercially qualified to do the work, costs can be controlled. There is nothing as costly to a project as having to remove a job from a contractor due to his lack of ability to perform.

It is a truism that when a contractor begins to lose money on a job, the quality of the work suffers and the demands for extras increase. While good inspection provides a defense, in many cases the contractor is paid for questionable extras in order to avoid litigation or delaying the total job.

Project Timing Control

Early in the project, timing is controlled by controlling the engineering effort to assure that bids are sent out, evaluated, and awarded in accordance with the project plan of execution. Once bids are awarded, the inspector and/or expediter has the primary responsibility to determine if work is progressing according to schedule. This requires that a schedule be worked out with the successful bidder (preferably before bid award). When delays are spotted, meetings with the contractor are required to determine a plan of action to return the job to schedule. On occasion it may be necessary to appeal to higher levels of management in both com-

panies. The sooner a problem is spotted, the better the chance that corrective action can be agreed upon.

Project Execution Format

Every project, from a simple single-well hookup to a complex offshore field development, goes through the steps of project management and control previously described. For simple projects many of these steps may not even be written down or formalized in any way, and drawings and specifications can be sketched on the back of an envelope. However, project engineers need to at least think about each item and consider how they are handling it if they are to do their jobs correctly.

As the project becomes more complex, more and more of these items must be formally included. This is true no matter whether the tasks are performed by the operator's staff, engineering consultant, or contractor. The project must progress from one step to the next; engineering, procurement and inspection must be accomplished; and the cost of performing these functions must be borne by the project.

There are several different project execution formats that must be considered in developing the project plan of execution. The choice of a specific format will determine which of these functions will be performed by a particular organization. Although there are almost as many formats as there are projects, most can be separated into the following four basic types which we shall call turnkey, negotiated turnkey, modified turnkey, and cost plus.

Turnkey

The turnkey format is used when the work is not completely designed. The conceptual study and project definition are normally complete. In this case the scope of the contractor's work would include the design engineering, detail engineering, procurement, inspection and expediting, startup manuals, and dossier, as well as the fabrication and possibly the installation of the completed facility. The following advantages are claimed by proponents of this format.

1. The project cost is established before work starts.
2. A single point of responsibility is established.
3. The contractor can design to take advantage of his construction efficiencies.

4. The contractor can speed up equipment delivery by performing engineering in such a manner as to get long lead time equipment on order sooner.
5. The contractor assumes risk.

The problems experienced with this format are:

1. Owner loses design control or can exercise it only at high cost in extra work.
2. Competition is limited to select firms with total design and construction capability.
3. Most large firms have different engineering and construction profit centers. Many large contractors bid the work to outside engineering firms. In either case, the contractor's engineers may be no more or less aware of construction efficiencies than a third party.
4. Because of the risks assumed by the contractor, he must add a contingency factor to his price.
5. By necessity there will be many subcontractors furnishing individual items of equipment to the turnkey contractor. Other than initial approval, the owner has no control over subcontractors.
6. The contractor's and owner's interests are not identical. The contractor has an incentive to provide the least costly quality for the fixed price, and does not profit or lose nearly as much as the owner with timely delivery. Therefore, the owner must provide inspection and expediting. It is very difficult to do this for subcontractors where no direct commercial relationship exists.

Negotiated Turnkey

The negotiated turnkey format recognizes that before the detailed engineering is complete, it is difficult for a contractor to provide a fixed price for the work while maintaining adequate owner control. In this format, the design and detail engineering are performed (normally for a fixed fee), so that long delivery items can be placed on order prior to completion of detail engineering. Once the scope of work is defined a turnkey price is negotiated.

The advantages claimed for this format are the same as those for the turnkey format with the added advantages of maintaining owner control and reducing contractor's risk and therefore the contingency factor in his

bid. The disadvantages are identical with the added disadvantage of eliminating much of the owner's leverage when it comes to negotiating the final contract.

Modified Turnkey

In the modified turnkey format, each work item is separated and bid turnkey as the scope of that work item is defined. In the previous two formats, the prime contractor does this in bidding out items of equipment to his subcontractors. The difference in this format is that the owner or his consultant do the bidding, awarding, and expediting. In addition, those items that are "sole sourced" to the contractor's construction arm in the two previous formats must be bid and evaluated. The main advantages claimed for this format are:

1. Control of the project is maintained.
2. Competition is maximized as individual work items can be bid to firms specializing in such work.
3. Owner's inspectors and expediters have a direct commercial relationship to all suppliers.
4. Contractor's risk and contingency is controlled to the extent desired by owner. For example, scope of work contingencies can be eliminated while weather contingencies are included.

The disadvantages with this format are:

1. Increased coordination of the contracts is required by either the owner or his engineering consultant.
2. The owner or his consultant must develop and monitor the plan of execution rather than this being a function of the turnkey contractor.
3. Project management costs are explicitly determined and not hidden in contract cost.

Current project management trends call for developing fields quickly for low capital and operating costs. In general, the priorities are: speed, initial capital, operating cost, and final capital, in that order. The main emphasis is to minimize the time between capital expenditures for facilities and the start of production (cash inflow). Achieving this aim can vastly improve rates of return for investments.

A modified turnkey format that uses a project management consultant with a separate engineering contractor selected by low bid may not be

readily applicable for fast-track projects. The more flexible approach of selecting an engineering consultant who is skilled at project management is often found to be useful. The engineering/project management consultant performs all design and project management work and bids equipment, fabrication, and construction.

The engineering company works as consultant to the owner at an early stage to define the scope of work. Once this scope is defined, the company can perform engineering work that is targeted toward completing the equipment packages and other long-lead bids. Engineering for auxiliary items, instrument bids, and offshore installation and hookup is performed at a later date. In addition to achieving fast-track results, this approach gives the owner a complete understanding of the work involved in the project and the associated costs. Above all, the strategy maintains flexibility at all stages as the project advances, allowing personnel to respond to new information that may develop during the project cycle time due to data from new wells, new reservoir analyses, negotiations with other operators, purchase of oil and gas, etc.

The advantages of this approach are:

1. Adequate communication between engineering and project management personnel
2. Low total costs
3. Quick deliveries
4. Minimal owner manpower requirements
5. Quick evaluation and response regarding project changes
6. Above all, flexibility to allow the owner to realign goals as new data become available.

On the downside, engineering costs may appear high because they are a specific line item and are not hidden in the cost of equipment.

Cost-Plus

The cost-plus format requires the contractor to be reimbursed for all direct costs plus a percentage of his costs for overhead and profit. Typically, this format is used where risk is high, or when there is insufficient time to solicit firm bids. Such a case would occur if construction were required within an operating plant, if it were necessary to repair storm damage, or if a simple field routing job were envisioned. The major dis-

advantage of this format is that the owner bears the risk of inefficient labor and job organization.

Comparison of Formats

The type of project format to employ depends on the nature of the project, the type of contractors available and their competitive position, and the priorities of the owner. There is no one answer as each project is different, and competition and priorities change from time to time.

One of the authors had the opportunity to be involved in two similar offshore projects for the same owner, which occurred almost simultaneously. Each was purposely set up in a different format to test the validity of the claimed advantages and disadvantages for this type of project at a particular point in time. Figure 13-22 summarizes the results of the comparison. It can be seen that both the cost and total elapsed time for the modified turnkey format were better than that of the turnkey format.

Figure 13-23 shows the results of another project done at the same time, which was set up in the negotiated turnkey format. When the negotiated price appeared too large, the format was changed to modified

	Project A Turnkey[1] ($M)	**Project B Modified Turnkey ($M)**
COSTS		
Production skids	$1,000	$875
Oil pumps and LACT	150[2]	170[3]
Power generation and heat	525	440
Installation	200	190
Offshore piping, electrical and instrument	500	515
Engineering	275	115
TOTAL COMPARED COST	$2,650	$2,305
TOTAL PROJECT COST	$3,065	$2,788
TIME		
Elapsed time from conceptual study	18 mo	16 mo

[1] *Breakdown furnished by turnkey contractor*
[2] *Three pumps*
[3] *Five pumps*

Figure 13-22. Turnkey vs. modified turnkey comparison.

turnkey. The original contractor bid on certain phases of the job and this total is shown as well as the actual cost when each phase was awarded to the low bidder. It can be seen that both the low bid submitted by the contractor when faced with closed bid competition and the actual cost were significantly less than what the negotiated price would have been. By comparing "actual cost" to "contractor low bid" it can be seen that a 10% savings in total project cost was realized by opening competition to firms other than the original turnkey contractor.

	Owner Original Estimate	Contractor Original Estimate	Contractor Negotiated Price	Contractor Low Bid	Actual Cost
Major equipment	$1,007,978	$691,473	$975,666	$716,031	$716,031
Material Fabrication		191,500	323,910	360,157	265,835
Load-out	5,000	8,670	7,200	4,833	4,833
Shipping and installation	205,000	151,814	357,100	300,575	200,036
Engineering		142,500	142,500	142,500	142,500
Contingencies	340,000	53,172	83,194	38,156	38,156
Common charges	283,515	374,331	376,843	374,176	374,176
Totals	$1,841,493	$1,613,460	$2,266,413	$1,936,426	$1,741,567

Figure 13-23. Negotiated vs. modified turnkey comparison.

APPENDIX A

Sample Project Assignment

Given:

1. A loosely consolidated, water drive sandstone reservoir in offshore South Louisiana.
2. Properties: Oil—30° API
   Gas—0.6 S.G.
   SITP—4,000 psi
   FTP (initial)—1,200 psi
   FTP (final)—100 psi
   Temp.—90°F
3. Production Forecast (Per Well):
   Q_o (initial) = 500 bopd
   Q_o (final) = 50 bopd
   Q_w (initial) = 0 bwpd
   Q_w (final) = 1,000 bwpd
4. Wells have a GOR at 1,000. When wells drop to low pressure and are gas lifted, they will have 350 Mcfd of gas lift gas per well.
5. Wells will eventually be gas lifted but not initially.
6. Gas sales pressure 900 psi.

7. Oil quality—1% BS&W. There will be no custody transfer at this facility, but oil will have to be pumped from an atmospheric storage tank into 1,000-psi operating pressure pipeline.
8. Water discharge, overboard—72 mg/l
9. Number of wells—10
10. Deck area is 7,500 ft

Problem:

1. Draw a process flow diagram for separation, oil treating, water treating, testing equipment.
2. Size all vessels and draw a separate sketch showing critical dimensions and internals for each one selected.
3. Show all pressure breaks on the process flow.
4. Draw mechanical flowsheet for well test equipment showing all lines, valves, and controls.
5. Size two lines in each of the following categories and pick a specific line size.
   a. Two-phase flow
   b. Liquid
   c. Gas
6. Submit all diagrams and calculations.
7. Schedule a conference at my convenience to review your results. This is your responsibility, not mine. I am your client.

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